JUN 4 1925

MECHANICAL ENGINEERING

INCLUDING THE ENGINEERING INDEX

Aristotle Was Right

Aristotle had the right idea. Change is everywhere in progress. All things seek an ever-fresh equilibrium..... We all have the sporting instinct. We like a good game. Where is there a game with goals such as you may win with research and engineering? Good service to recognized human needs is usually rewarded in the hive-tokens of wealth, but that is only the subsidiary, not the necessary criterion of happiness. Most pioneers seem to enjoy life as a healthy sport. Whether they foresee the value of their work or not, they don't need to be continually "cashing in." They are rich because they hold nature's notes. If you read the lives of men like Pasteur, Faraday, Darwin, Watt, Kelvin, Edison and Steinmetz, you will see that to have an intense interest and follow it is a game than which there is no greater.

WILLIS RODNEY WHITNEY

(Excerpts from Aldred Lecture at M.I.T.)

JUNE 1925

THE MONTHLY JOURNAL PUBLISHED BY THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

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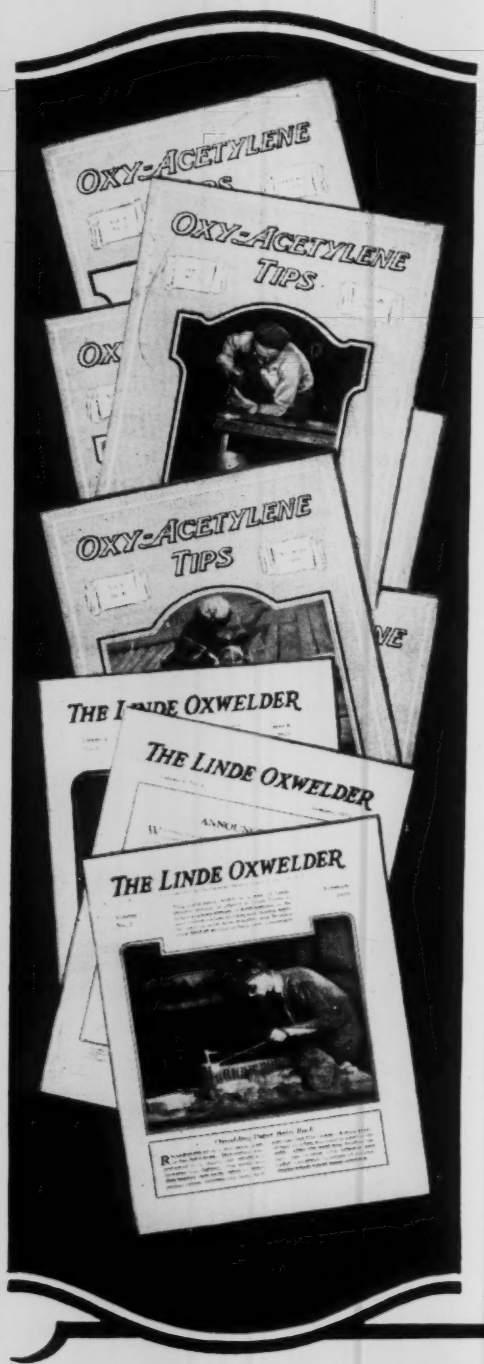

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Volume 47 June, 1925 Number 6

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A. M. Coyle

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MECHANICAL ENGINEERING

Volume 47 June, 1925 No. 6

The Oak Grove High-Head-Turbine Development of the Portland Electric Power Company

Particulars Regarding a Recently Installed 35,000-Hp. Vertical Pelton Turbine of the Francis Type Operating under a Head of 860 Ft.

By ELY C. HUTCHINSON,[1] SAN FRANCISCO, CAL.

BY FORCE of circumstance the development of machinery for the utilization of water power on the Pacific Coast has naturally inclined more toward the construction of water wheels and turbines for high heads than otherwise. The mountainous regions of the Pacific Coast, from Washington to Mexico, offer natural sites for high-head developments to an extent not existing elsewhere in the United States. The first developments from which have sprung the present-day high-head Pelton wheel, known under the generic term of "Pelton" throughout the world, were in California, where Lester Pelton, an old miner and millwright, is credited with the invention of the split bucket which, in a more improved form, is still being used in modern equipments.

The first turbines installed in the western country were of primitive construction, built from designs that were developed largely by cut-and-try methods, for installation under comparatively low heads.

Impulse turbines or Pelton wheels were used almost exclusively in the Pacific Coast country, in the larger and more important installations, until about fifteen years ago. At this time the growing paucity of accessible high-head sites for Pelton-wheel installations caused engineers to turn somewhat to the possible development of somewhat lower heads by the use of Francis turbines. A number of such turbine installations were made of a capacity considerably exceeding earlier designs and constituted a marked advance in the art of turbine development in the West. Numerous plants were installed for heads lower and water quantities larger than were suited to Pelton wheels, and economic conditions were responsible for leaving those sites offering the combination of high head with large water quantity in a largely undeveloped condition.

FIG. 1 GENERAL VIEW OF OAK GROVE PLANT WITH PENSTOCK AND SURGE CHAMBER

As a result of the growth of industry and increasing demand for power there naturally followed a consolidation of the smaller companies into operating systems of considerable magnitude and greatly enlarged resources. The public-utility companies formed by such consolidations were able to look upon the power problem and the development of added capacity to meet the existing and growing demands from a much different and more comprehensive viewpoint than had been possible in the earlier constructions.

The World War and its demand for the conservation of resources led to the interconnection of the large companies, forming a network linking together practically all of the power-producing sources on the Pacific Coast. It can be well understood, under such circumstances and with such background, that the best engineering thought would be led toward the comprehensive development of entire watersheds with their rivers and tributaries into single projects. The increased resources of the power companies and their superior ability to finance projects of greater cost made it possible to turn again to those power sites, the development of which in the earlier days had been beyond the means then available. The fact that such projects were generally more remote from the market no longer presented an insurmountable barrier, for long transmission lines and the use of higher transmission voltages eliminated this difficulty. However, with the advent of the long transmission lines it became necessary to give consideration to other factors in the design of hydroelectric machinery. It followed logically that the future power houses should contain units of such capacity that a single machine would be able to charge a transmission line of great length. The growth of the power demand, supplied by the larger companies, also made it necessary to consider individual units of large capacity which would operate continuously upon a block or base load and at efficiencies which would insure the most economical use of the water supply.

Large units would also make possible an economy in the general power-house installation that would be very desirable. It followed logically that water-wheel units of the Pelton type, of capacity sufficient to meet the general operating requirements, would be found unsatisfactorily large and of relatively slow speed if used under some of the heads that were available for development. Economically, therefore, the Francis turbine offered a very attractive possible solution for such installations, provided it could be made dependable and its operation and maintenance under heads hitherto untried could be successfully demonstrated.

To secure these results required a rather radical departure from the accepted low-head-turbine design, and in this connection the experience in operating high-head Pelton wheels has proven very valuable. It will be plain that any successful high-head-turbine design must avoid with great care not only the ailments of ordinary turbines such as unbalance from pressure differentials in and adjacent to the runner, wear and pitting of runner vanes, vibration and draft-tube disturbances at partial gate openings, penstock from shocks, etc. but the augmented effect of high velocities resulting higher heads. Then there were also the unknown effects resulting from high-head operation for which no precedent whatever existed.

[1] Vice-President and General Manager, Pelton Water Wheel Co. Mem. A.S.M.E.

For presentation at the Portland Regional Meeting of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, Portland, Ore., June 22–25, 1925.

The first distinctly high-head Francis-type-turbine design was developed for installation at the Kern River No. 3 Power House of the Southern California Edison Company. This plant uses the waters of the Kern River and consists of two 25,000-hp. vertical Pelton turbines of the Francis type, operating with a head of 810 ft. and with speeds of 500 or 600 r.p.m., dependent upon whether the plant is connected to the 50- or 60-cycle portions of the Edison Company system. This installation was described rather completely in an article appearing in MECHANICAL ENGINEERING, April, 1922. The installation has been in successful operation for about three and one-half years at this writing.

The Oak Grove Plant of the Portland Electric Power Company, which went into operation in July, 1924, is the second of the high-head installations, and so far as known is the highest-head plant in existence in which turbines are being used.

DESCRIPTION OF THE PORTLAND ELECTRIC POWER CO. SYSTEM

The Portland Electric Power Company has for many years supplied electric power to the industries in Oregon and has participated in the evolution of hydroelectric-plant design and operation as have few others in the western states. There is in service on its system at the present time a series of low-head reaction turbines installed in 1889 to operate under heads of from 20 to 40 ft.; each unit driving a generator of 425 kw. capacity. In 1907 a turbine was installed for operation under 130 ft. head and direct connected to a 2500-kw. generator. Later, in 1911, a third installation was made, utilizing a number of turbines operating under 81 ft. head and each direct-connected to a 3300-kw. generator. A year later, that is, in 1912, a further development was made utilizing a head of 325 ft. and a generator of 5250 kw. capacity. Therefore the Portland Electric Company has participated in the growth of the hydroelectric industry over a considerable period of years, during which the operating head for turbines on their system has changed from 20 ft. in 1889 to 860 ft. at the Oak Grove Plant in 1924, while capacities per unit have grown from 425 kw. to 25,000 kw. During this same period the system has grown until it now has a capacity of 85,730 kw., serving a population of 350,000 distributed over a total of 500 square miles. This undertaking requires 550 miles of high-tension transmission lines and over 9200 miles of distribution lines, the present yearly output being over 350,000,000 kw-hr. Here will be plainly seen the typical development and conditions responsible for the present-day large-capacity hydroelectric units.

The Oak Grove Plant (see Fig. 1) is situated on Oak Grove Creek, a tributary of the Clackamas River about 70 miles from Portland, Oregon. The stream flow is very uniform and dependable in dry seasons. Water is diverted about six and one-half miles above the plant by a concrete intake dam of arched form 68 ft. high and 168 ft. long. The first 1500 ft. of the conduit from the dam is a concrete-lined tunnel 14 ft. in diameter, this section being designed of sufficient capacity for carrying water for the full development which ultimately will consist of three 35,000-hp. turbines. A 9-ft.-diameter steel pipe continues from this point to a concrete Johnson differential surge tank at the crest of the hill from which the principal drop to the power house is secured (see Fig. 2). A wye or manifold branch is also provided at the lower end of the 14-ft.-diameter tunnel section for the connection of future pipeline conduit which will be installed when the additional units are put in. A penstock proceeds from the surge tank to the power house a distance of 1200 ft., tapering from a diameter of 8 ft. at the top to 6 ft. at the bottom, where the turbine is connected with a static head of 930 ft. Between the outlet to the surge tank and the upper section of the penstock a Johnson valve is installed.

The power house at the present time contains one 35,000-hp. vertical Pelton turbine designed to operate under an effective head of 860 ft. and driving a General Electric generator of 30,000-kva. capacity, wound for 11,000 volts at 514 r.p.m. A direct-connected exciter is mounted on the upper end of the generator, and the rotating element, consisting of generator, rotor, and turbine runner, with their respective shafts, is supported by a Kingsbury thrust bearing. The generator is ventilated by means of closed air ducts which surround it, and air circulation is induced by the rotation of the machine itself. Provision is made for forced ventilation if required. (See Fig. 3.)

THE HYDRAULIC TURBINE

As previously stated, the turbine is of the vertical type. Its shaft is rigid-coupling-connected to that of the generator above; there being in all three steady-bearing supports for the entire shaft length. Two of these are incorporated in the generator and the third is in the turbine. The runner is mounted upon a tapered portion of the shaft which extends below the turbine steady bearing.

The main operating floor of the power house (see Fig. 4) is located at the upper level of the generator. The governor and the control for the main shut-off gate are located on this operating floor and suitable extensions are continued downward to the turbine operating deck two stories below. Still further below the turbine operating deck is the draft-tube deck, from which access is had into the turbine casing by removal of certain of the draft-tube sections which are made of metal and specially designed for the purpose. The draft tube is of the Moody spreading type, formed in the solid concrete substructure of the power house. The turbine casing is made of annealed cast steel in several sections, heavily bolted together with high-pressure gaskets (see Fig. 5). Extension pads are provided on the casing sections for the support of the servomotor cylinders. The upper and lower ring covers for the turbine casing are made of cast iron and both have bronze-fitted bearings for supporting the turbine guide vanes. The lower ring casing cover is made in two sections, so constructed that it will be unnecessary to disturb in any way the assembly of the turbine guide vanes when removing the turbine runner. By shortening the time out of service when changing turbine runners, this feature of design is of great importance and has been found of extreme value.

The runner is a single bronze casting having a capacity of 35,000 hp. at an effective head of 850 ft. The normal operating speed is 514 r.p.m., and the specific speed (English system) is 21.5. Very complete arrangement is made for removing the runner and putting it back into place in the shortest possible time. A special portable jack, which connects rigidly to the shaft, is used to force the runner on or off.

The upper portion of the draft tube is made in two sections of cast iron, each being fitted with roller-bearing car wheels. The lower cast-iron section is connected to a collar piece or thimble by means of a calked and leaded joint of the bell-and-spigot type. This joint is quickly disconnected by removing a sectionalized flange or bell portion. The draft-tube sections are then lowered to rails set into the floor, after which they are quickly rolled out of the way. All interior parts are handled by the power-house crane with a cable passing through the hollow shaft. The upper draft-tube section acts as a carriage for the turbine runner; the runner being lowered to it and then secured by means of special screws provided for the purpose; after which both runner and the upper tube section are lowered as one piece to the rails previously mentioned.

The runner seal rings are rubber lined and arranged for water lubrication by means of an external connection directly from the penstock. The stationary rings are fastened to the casing cover plates and are the ones fitted with the rubber lining. An allowance of 0.01 in. in the diameter is provided between the stationary and rotating portions for runner clearance. This is of course an extremely small allowance, but by the use of rubber seal rings no greater allowance is necessary as water lubrication will amply protect the wearing surfaces, and any slight out-of-alignment condition between the rotating turbine runner and the stationary seal rings will act only to compress the rubber ring at the point of contact, without damage.

The shaft is fitted with a wearing sleeve at the point where it passes through the stuffing box, and the stuffing box is itself of interesting and unusual construction.

Turbines designed for lower heads use no stuffing boxes at all at the point where the shaft passes from the interior of the casing, a grease seal being normally considered ample protection. Obviously, however, should any internal hydraulic condition occur in the high-pressure constructions which would permit the building up of pressure on the top of the turbine runner, the result would be a discharge of water at the point of shaft entry under high-pressure conditions. As this is to be avoided, a stuffing box packed with a specially devised graphite-metallic packing has been provided. To prevent generation of excessive heat by the rubbing of

this packing against the shaft at the high rotating speed at which the unit runs, it is of course desirable that the packing be kept as loose as possible. To prevent any leakage water passing the stuffing box at all under such conditions, a lantern gland is introduced between packing rings, thus forming a chamber into which any seepage water passing the first rings of the packing may be collected and carried away to tailwater.

As a further safeguard, however, a water-operated ejector is furnished to remove any seepage from the guide-vane or shaft stuffing boxes.

The main steady bearing is of the babbitted and oil-lubricated type, made in halves and supported directly from an extension of the upper ring casing cover. It is arranged for pressure lubrication introduced at the center, and overflows are provided both top and bottom which return the oil to a sump whence it is raised to a storage tank some distance sbove. Means for collecting the oil from the bearing have been carefully worked out so that there is no seepage or disagreeable accumulation of oil outside. A check system of oil grooving is provided in the bearing, each set of grooves being fed from circular channels which are kept constantly filled by the pressure oil.

The turbine guide vanes (see Fig. 6) are made of annealed cast steel and are of the disk type. This form is particularly useful in high-head-turbine construction as, by interposing metal disks between the spindles and the guide vane proper, it is possible to secure a very rugged connection between the preferably slender shape of the guide vane and the necessarily large diameter of the spindles which move it. Were the disks omitted from the guide vanes, it would be necessary to shape them very differently, making them thick at the center where the spindles attach and seriously affecting the hydraulic form. The full width of the guide vanes for the Oak Grove unit (which, it will be remembered, is of 35,000 hp. capacity) is but $7\frac{1}{2}$ in. Their supporting spindles rotate in grease-lubricated, bronze-bushed bearings. The unbalanced thrust, or upward lifting tendency of the guide vanes (the ends of the spindles being exposed to the casing pressure), is carried on specially constructed thrust rings, grease-lubricated, and made of Parsons' metal. The stuffing boxes through which

FIG. 2 PORTION OF THE 9-FT.-DIAMETER RIVETED-STEEL FLOW LINE

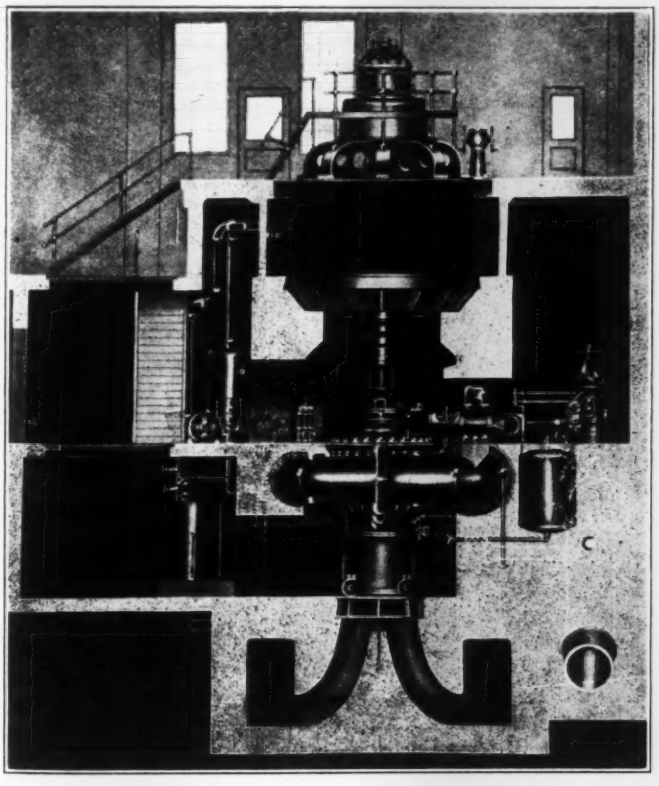

FIG. 3 POWER-HOUSE ARRANGEMENT OF GENERATOR, TURBINE, DRAFT TUBE, AND AUXILIARY EQUIPMENT

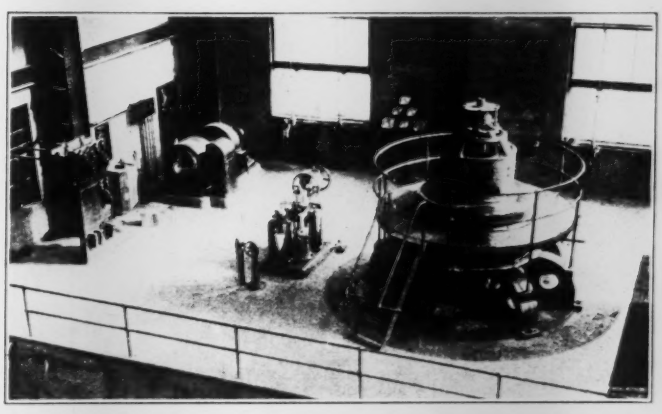

FIG. 4 MAIN OPERATING FLOOR OF THE OAK GROVE PLANT

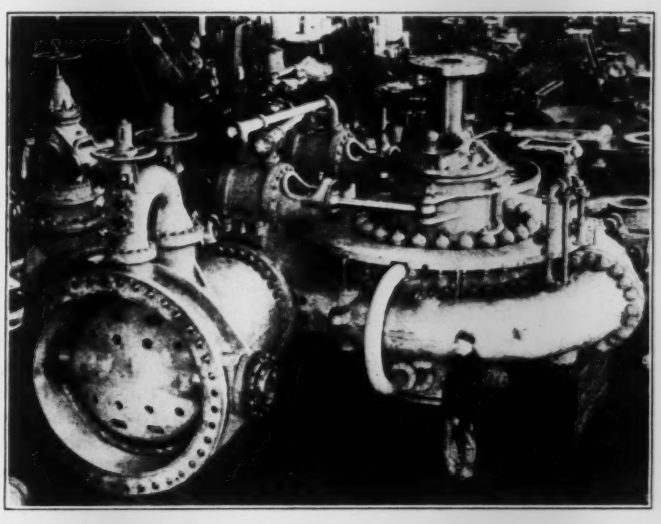

FIG. 5 35,000-HP. PELTON REACTION TURBINE: SHOP ASSEMBLY

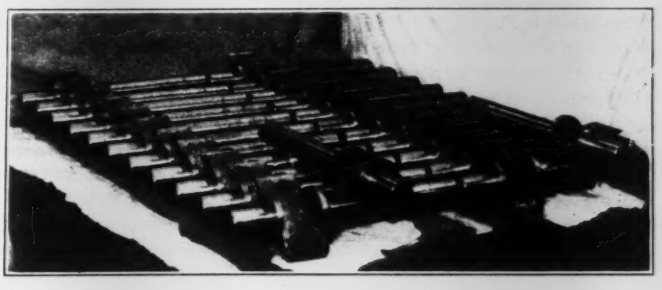

FIG. 6 DISK-TYPE GUIDE VANES FOR OAK GROVE TURBINE

the guide-vane spindles extend for their connection to the gate-operating mechanism, are extra long and provided with lantern glands and drains to carry off seepage water in the manner previously described in connection with the stuffing box for the main shaft. Outboard bearings support the extreme ends of the guide-vane spindles.

The speed-control mechanism follows the usual plan for large high-capacity turbines, excepting perhaps that instead of mounting the servo-motor cylinders in the wall of the concrete pier which supports the generator, they are bolted and doweled directly to finished surfaces provided for the purpose on the cast-steel spiral

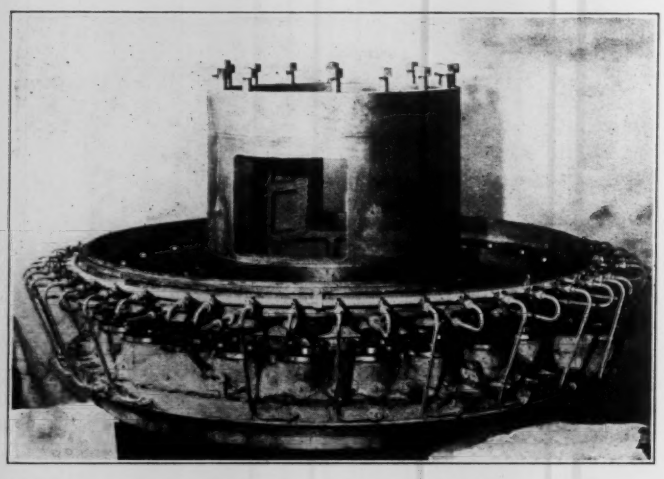

FIG. 7 UPPER RING COVER FOR TURBINE CASING SHOWING PRESSURE LUBRICATING SYSTEM FOR GUIDE-VANE BEARINGS

casing. This construction makes it possible to completely assemble the servo-motors with their crossheads, connecting rods, and shifting ring, including its connections to the guide-vane levers, as a part of the work of manufacture, and to actually try out all of this mechanism in the shops under pressure prior to shipment. The complete alignment and assembly of all of the speed-control mechanism in the manufacturer's shops is of great advantage since it assures perfect alignment and simplicity of assembly at destination. It also safeguards against interference or errors of dimension at a time when such difficulties may be easily and quickly corrected with facilities not available at the power house. The servo-motors are of cast iron, bolted and doweled to the turbine casing and enclose snap-ring-packed pistons. External crosshead guides very similar to those used in steam engines provide against any possibility of binding or distortion from angular stresses from the connecting rod. The connecting rods are forged from steel, turned and polished, and provided with bronze-bushed bearings.

The shifting ring which receives its motion from the servo-motors is made of cast steel and makes its rather short arc of movement around a circular, central column cast integral with the top ring casing cover. Cast-iron links, bronze-bushed and with specially reduced section to break in advance of any other portion of the gate mechanism, are fastened to one end of the shifting ring and at the other to cast-steel guide-vane levers. Tapered keys are used for connecting the outboard ends of the guide-vane spindles and the gate levers. The machining of the guide vanes is done entirely to gages, which makes it possible to assemble the entire guide-vane system without previous marking or adjustment. Spare guide vanes can also be put into place without the necessity of marking and cutting the keyways afterward.

Water leaking past the upper seal rings is prevented from accumulating on the back of the turbine runner and thus causing a pressure which would greatly increase the load on the thrust bearing, by a set of exterior pipe connections which span the turbine casing and connect the chamber above the runner directly into the draft tube, securing thereby an ejector effect. By this arrangement it is possible to maintain little or practically no pressure above the turbine runner. Users of turbines where adequate means have not been available to remove this pressure will recognize the distinct advantage of this construction.

The lubrication system is carefully worked out and is both complete and simple. The guide vanes, both top and bottom, are lubricated by a manifold system from a central, hydraulically operated, compression grease gun of large capacity (see Fig. 7). The steady bearings are lubricated by oil under gravity pressure from a tank located well above the highest bearing of the generator. An oil pump, of the rotary type, belt-driven from the turbine shaft, delivers the oil discharged from all the bearings of the unit from a sump tank located at one side of the turbine into the storage tank above. An oil filter is provided for clarifying the oil for all the bearings. In addition to the belt-driven rotary pump, an auxiliary motor-driven pump of the same size is also provided. This is used to start the lubricating system before the main turbine unit is running, and also comes into service if the belt-driven equipment is shut down. The rubber seal rings are lubricated both by the water leaking through them under natural conditions and by an external water supply taken directly from the penstock as previously described. This makes it possible to turn a supply of lubricating water into the seal rings before the unit is started up and also during the time that the unit is shutting down and the guide vanes have

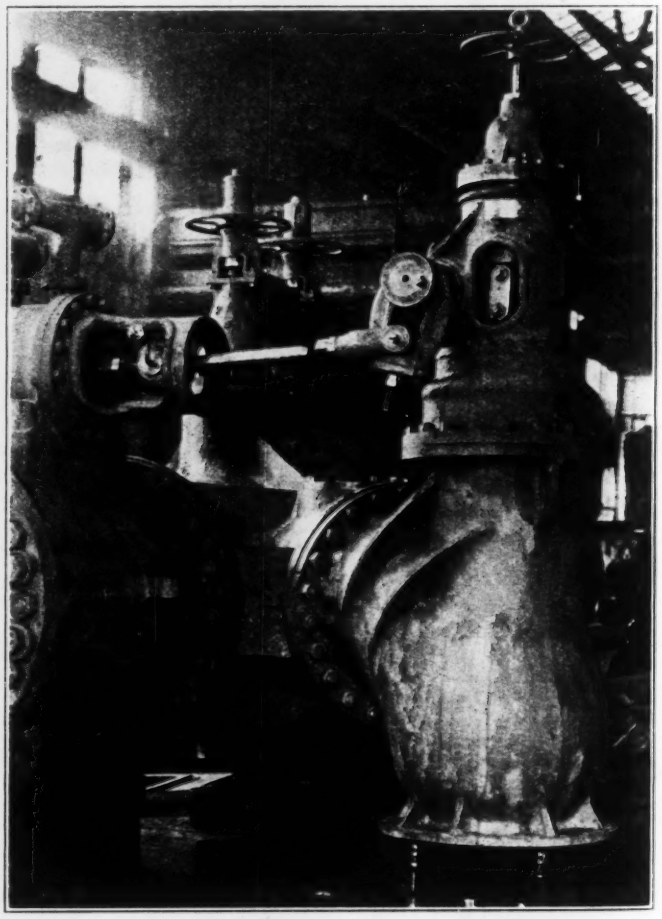

FIG. 8 TURBINE RELIEF VALVE SHOWING MECHANICAL CONNECTION TO GATE-SHIFTING MECHANISM

been closed by the governor. Under such conditions the necessary lubricating-water supply to the seal rings would be cut off were this auxiliary means not provided.

THE RELIEF VALVE

The pressure regulator, or relief valve, provided for the Oak Grove turbine is of particular interest to those more familiar with the types of regulator used on turbines of moderate head, although to users of impulse or Pelton wheels it will not appear very different from the auxiliary relief, needle regulating nozzle used in connection with the modern Pelton wheel. In the design of the relief valve it was necessary to provide a construction which would have characteristics of discharge in agreement with the flow of the water through the turbine guide vanes, since if a change of quantity flow through the turbine was not met with the relief of an equal

quantity through the relief valve during the periods of speed regulation, a difference in velocity in penstock flow would result. Under high-head conditions the danger from variations of penstock velocity may easily be more serious than with turbines operating under ordinary heads. It was also very necessary to secure a smoothness of operation and a freedom from vibration in the relief-valve discharge as the high-head conditions would easily make disturbances of this sort a serious menace. The disposal of the jet discharging from the relief valve and the effective quenching of its energy is ordinarily accomplished by discharging directly into a circular baffle of vortex form as used ordinarily for high-head Pelton-wheel installations. In the present case, however, the jet is turned through an arc slightly in excess of 90 deg. by means of a substantial cast-iron pipe elbow and is allowed to discharge directly into the stream below the power house in a spectacular but harmless fashion without disturbing the draft-tube tailwater.

The relief valve (see Fig. 8) is of the needle type and has a rigid mechanical connection to the gate-shifting mechanism of such strength that should an obstruction prevent its opening it will effectively block any further movement of the entire gate system. This is an essential point of design as it is obvious that it is much better to stop all movement of the governing mechanism than to have a breakage which would permit a part of it to function independently and without the very necessary coördination of the remainder.

An oil-cataract dashpot insures free movement of the relief valve in the opening direction, while requiring a slow movement of the plunger in the direction of closure. Valves are also provided in the dashpot structure through which the period of closure may be adjusted. Also, no movement at all will occur for motion of the main-turbine guide vanes not sufficient to endanger the pipe line. The relief valve can also be adjusted to act as a continuous bypass or a partial bypass. In the latter case the valve may be made to close slowly to a point of opening previously adjusted to remain constant.

As it is very difficult to determine the exact amount of servo-motor stroke which will produce the full-load capacity of the turbine, an adjustable lever is provided on the relief valve by which it may be set to secure the full-opening stroke after assembly and test at the plant. This feature has proven both desirable and useful, and has been the means of securing maximum safety to the plant.

The discharge leading from the relief valve is ventilated to prevent any tendency of the discharge water to remain in the pipe elbow. This insures the maximum discharge value from the valve.

The Butterfly Valve

Should the turbine guide vanes become worn or bent the leakage through them under the high head might easily be sufficient to rotate the unit at higher than synchronous speed under no-load conditions. It has therefore been necessary to provide means whereby the flow of water into the turbine may be regulated, if need be, independently of the guide vanes in order that the unit may be easily synchronized with others of the system.

For this purpose a large butterfly valve is connected directly to the inlet of the turbine casing. This valve is made of cast steel throughout and the disk is provided with hydraulically operated snap rings seating against a stationary bronze face so that it is practically watertight under the full static head of 930 ft. This butterfly valve operates under a higher head than ever previously used and its construction is extremely rugged. The disk, or gate proper, is mounted upon a forged nickel-steel stem and is moved with very substantial worm gearing by means of a reversible Pelton water wheel which is controlled from the operating floor two stories above. The water wheel is powerful enough to operate the disk under all conditions of flow from zero to maximum with the turbine gates wide open. An electric-motor auxiliary is also provided for operating the gate when there is no water in the penstock and also when the pressures are balanced by means of a bypass. Limit stops are provided for both the water wheel and the motor which prevent overrunning at either end of the main-disk movement.

Inasmuch as the control for the butterfly valve and its bypass is on the main operating floor two decks above, an indicator is provided in the form of a miniature cross-section of the valve itself. This miniature has a movable butterfly disk within it which takes the exact position of the main disk at all points of movement; thus the relation of the main disk to the valve body and point of final setting are always clearly before the operator.

The Governor

The governor (see Fig. 9) is also mounted on the main operating floor and its oil piping as well as drive connections are extended vertically downward as required. The governor is self-contained with an enclosed-type centrifugal element and all of the usual accessories required for the speed control of a modern, high-capacity installation. Because of the high operating head at the Oak Grove plant, it will be realized that a comparatively small water quantity is representative of a large horsepower difference. This may be emphasized by pointing out that the total servo-motor stroke representing the difference between "no load" and "full capacity" of 35,000 hp. will not exceed 5 in.

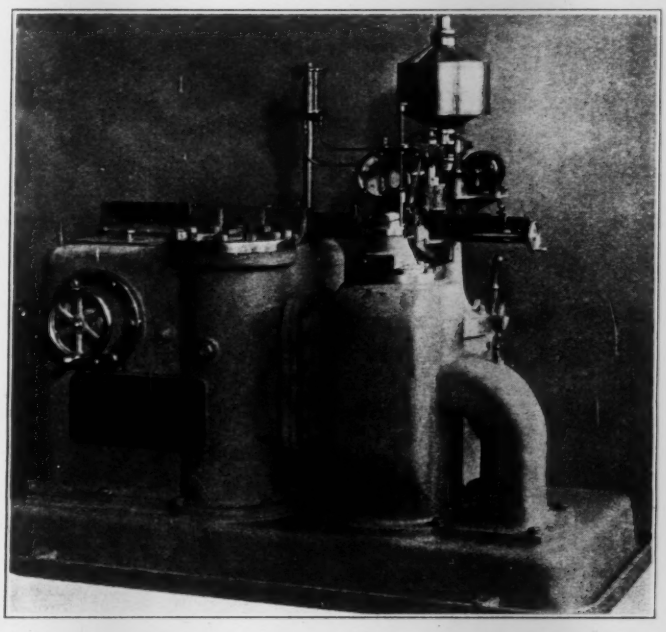

FIG. 9 PELTON-TYPE D. G. ACTUATOR WHICH REGULATES THE TURBINE SPEED

Delicacy of control and sensitiveness of the entire speed-governing mechanism are plainly an essential characteristic under such circumstances. While possessing the quality of extreme sensitiveness for speed-regulating purposes, such extreme sensitiveness would, however, make it difficult to obtain the steady movement necessary when it is desired to synchronize the unit and it is operating at no load. The governor mechanism is specially designed to meet these requirements. The synchronizing operation is handled entirely from the switchboard by means of a simple push-and-pull reversing switch which operates a synchronizing motor on the governor.

An easily adjustable load-limiting device is also furnished. Protection from accident in event of the governor belt's breaking is provided and an emergency solenoid shutdown device is provided. This is push-button-controlled from the switchboard and may also be connected to various other parts of the station so that the entire unit may be automatically shut down on the development of any difficulties whatever, whether they be of an electrical or hydraulic nature.

The operation of the governor in changing from hand to governor control or vice versa is extremely simple as it requires but the movement of a single-throw lever, all other adjustments or valve operations taking place within the governor itself.

An entirely independent oil jacking-pump is provided for the unit by means of which the gate-shifting mechanism may be moved for its entire stroke in either direction when the plant is shut down and no pressure oil is available.

INDIVIDUAL OIL-PRESSURE SET

The unit is provided with an individual oil-pressure system (see Fig. 10) having duplicate oil-pressure pumps, one driven by an electric motor and the other by a water motor. Each of the succeeding units to be installed in the power house will also be furnished with its individual oil-pressure system. The two pumps, each of which is of a sufficient capacity for the full governing requirements of the turbine, connect to a single air-pressure accumulating tank through suitable piping and unload through a single automatic volume-controlled unloading valve. The volume-controlled operation is used in preference to the pressure-controlled system as it avoids any possibility of losing the oil. With the pressure-operated systems it is possible that an excess accumulation of air in the pressure tank may force the oil level downward until a series of servomotor strokes would completely exhaust it and cause serious

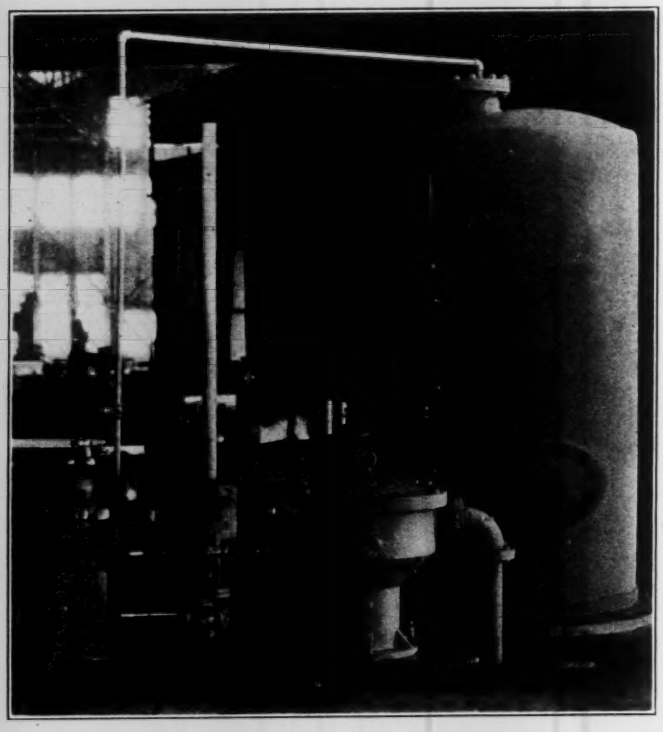

FIG. 10 OIL-PRESSURE SYSTEM FOR GOVERNING; ASSEMBLED FOR SHOP TESTS

damage. Under the volume-controlled system the oil is maintained at a constant level within the tank and an excess accumulation of air under similar conditions will only result in an increase of pressure and not a reduction of oil volume in the tank. Should the pressure of air exceed a predetermined maximum it will discharge automatically through a relief valve provided for the purpose. A sufficient amount of air to make up losses in the system may be pumped continuously through the oil pumps, making it unnecessary for the station operator to give this detail any more than general attention. Oil for the pumps is drawn from a sump tank below the floor into a self-priming strainer so constructed that after starting for the first time the pumps will always be primed regardless of the period of their shutdown.

THE JOHNSON VALVE

At the upper end of the penstock just downstream of the surge tank a Johnson valve is provided. This forms the main emergency-shutdown control for the entire unit and is arranged so that it can under emergency conditions be shut down by means of a press button at the power house. It cannot, however, be opened at the power house, this operation being accomplished only at the valve itself. To further improve the safety conditions the shutdown control switch at the power house is enclosed in a metal box having a glass front. The shutdown emergency operation therefore requires the breaking of this glass, and the physical act of pressing the button for the express purpose of shutting down the valve. Cases have occurred where the shutdown control intended for emergency purposes has been accidentally pressed and the results have been serious. Likewise there have been experiences where accidental opening of the emergency control valve has occurred with men at work on the turbines. Neither of these conditions can happen under the arrangement described. As a further safeguard, the Johnson valve is provided with an automatic closing arrangement which will come into action if the velocity in the penstock becomes approximately twice normal; such a plan being well worth while when the potentialities are considered.

CONCLUSION

In concluding, it is safe to say that the operating experience at Oak Grove and the more than three and a half years of operation of the Kern River No. 3 Plant of the Southern California Edison Company have successfully demonstrated the fitness of Francis-type turbines for high-head installations. With these plants as a background we shall move forward to bigger and greater things in turning our natural resources to the benefit of civilization.

The author wishes to acknowledge the assistance of Mr. E. D. Searing, Advisory Engineer of the Portland Electric Power Company, in the preparation of this paper.

DATA of a series of tests begun in the Laboratory for Testing Materials at the University of Wisconsin in February, 1911, under the direction of Prof. M. O. Withey. These tests were conducted over a period of about 12 years, which period, in view of the accelerated action, was sufficient to reveal the results which might be expected to obtain under ordinary conditions over longer periods. During this time the effects of such variables as alternate wetting and drying, salt in the mixing water, salt in the water bath, richness of mix, and thickness of covering on the rods were studied.

Tests show that rusting of bars imbedded in concrete will occur to some extent where oxygen and moisture are present, but only after a long period of time and in mixes lean in cement. No rusting of any kind was noted in the fresh-water specimens until the seven-year tests, and this occurred on specimens of 1:2:4 and 1:3:5 mixes. The porosity of the rich mixes is considerably less than that of the leaner mixes; therefore in a given time the amount of oxygen and moisture that can reach rods imbedded in a rich mix is less than can penetrate to rods similarly imbedded in a lean mix. Usually rusting occurred at the air pockets underneath the rods. Where chemicals such as salts came in contact with the rods the rusting action was greatly increased, and, in accordance with the above conclusion, the greatest increase was noted in the lean mixes. As was expected, rusting occurred first in the specimens $1\frac{1}{2}$ in. thick because oxygen and moisture could reach the rods in these specimens most easily. Since there was no marked rusting of any of the fresh-water specimens, whether they had been soaked in sea water or not, it is probable that the effect of the short time treatment in the sea water was negligible.

The cycle of immersion and drying which was carried on over the first five years of the test was probably more severe than would be encountered in practice, but this accelerated action was probably of no greater extent than that which would occur under actual weathering conditions extending over a much greater period of time. During the drying it is probable that the forces exerted on the particles of cement and aggregate by the crystallization of the salt broke down the minute cell walls and thus increased the porosity of the concrete. This increase in porosity permitted a freer access of oxygen and moisture, which, as stated above, subsequently caused greater corrosion of the rods.

However, in no case was the corrosion so deep or the deterioration so complete that the capacity of the rods to carry static loads was materially decreased. On the other hand it is probable that the resistance of the rods to impact or repeated loads was materially lessened because of the concentration of stresses which would occur at the bottom of the pits under such loadings. After longer exposure to such action even the static capacity of the rods might be decreased.

These tests show conclusively that salts, and chemicals having like action on steel, should be avoided in the mixing water for reinforced concrete. *The Wisconsin Engineer*, March, 1925, pp. 99–100 and 112)

The Increase in Thermal Efficiency Due to Resuperheating in Steam Turbines

By W. E. BLOWNEY[1] and G. B. WARREN,[1] SCHENECTADY, N. Y.

The results of a large number of tests on turbines at different superheats have been analyzed to form a basis for calculating the gain in thermal efficiency due to resuperheating. These results show that the heat consumption of a turbine installation may be decreased from 6 to 7 per cent as a result of resuperheating the steam. The probable gain due to more than one resuperheating has been obtained. These results indicate that, when taking into account the pressure drop in the resuperheaters, the gain possible by going to more than two stages of resuperheating would be very small. The effect of resuperheating upon the exhaust conditions and capacity of the turbine is discussed. Curves are presented to show that the gains due to the resuperheating and the regenerative cycles are very nearly independent of each other and will be nearly additive.

THE purpose of this paper is to analyze available data relative to the increase in turbine efficiency with increased initial superheat and to determine therefrom the probable gain due to resuperheating in high-pressure turbine installations. The resulting resuperheat cycles have also been analyzed in two instances in combination with regenerative or steam-extraction feed-water-heating cycles.

It has been recognized for a long time that whatever gain would result from resuperheating was in the main part due to the increase in the steam-turbine efficiency resulting from the greater superheat and reduced moisture content of the steam in the turbine, rather than to the thermodynamic improvement of the heat cycle.

The data relative to the gain in turbine efficiency with increased initial superheat are not as complete as might be desired. In the main it has been necessary to base the present series of calculations upon tests on turbines of from 2500 to 10,000 kw. capacity, but very complete tests are available at low initial pressures and fairly complete tests are available at moderately high pressures at these capacities. The data seem to point to the conclusion that the slope of a superheat-efficiency curve is almost independent of the type of turbine.

The superheat-efficiency curves available have been plotted in comparison and graphically averaged to give curves of representative slope; these average curves have then been analyzed on such a basis as to make the results available for determining the increase in the turbine efficiency due to resuperheating.

Definitions

In this paper improvements in the heat consumption of a turbine will be spoken of as "percentage improvements in the thermal efficiency."

The following terms are used as defined:

Thermal Efficiency of Turbine: The ratio between the heat equivalent of the output of the turbine at a prescribed point and the total net heat input into the turbine.

Turbine Efficiency: The ratio between the heat equivalent of the output of the turbine at a prescribed point and the adiabatic available energy between the pressure and temperature limits through which the turbine works.

Turbine Output: Unless otherwise specified, the turbine output has been assumed to be the output at the turbine-wheel hubs and the efficiencies have been based on this value. The losses due to the bearings, high- and low-pressure packings, generator, etc. have thus been excluded and the leaving velocity loss has been assumed to be zero, since these vary with each installation and design.

Fraction of Adiabatic Drop at Which Resuperheating Takes Place: This value is the ratio of the adiabatic drop which takes place before resuperheating as compared to the total adiabatic heat drop from the initial to the final pressures along the initial entropy line.

[1] Turbine Engineering Dept., General Electric Co. Jun. A.S.M.E.

Contributed by the Power Division and presented at the Annual Meeting, New York, December 1 to 4, 1924, of The American Society of Mechanical Engineers. Abridged.

Assumptions Used in Analysis of the Gain in Turbine Efficiency with Increased Initial Superheat

Curve *A*-3, Fig. 1, shows a typical turbine-efficiency-initial-superheat curve. The improvement in the turbine efficiency as the superheat increases is due to an improvement in the individual stage efficiency and to an improvement in the reheat factor R with increasing initial superheat.

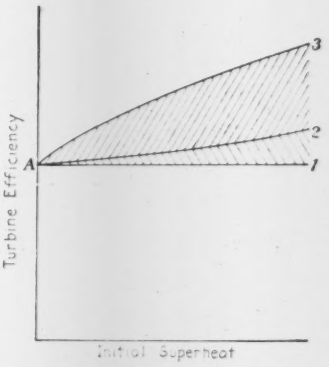

Fig. 1 Typical Superheat Turbine-Efficiency Curves

A modern turbine is made up of a number of individual stages, each stage taking a part of the pressure drop, and thus a part of the energy drop. Since each stage is less than 100 per cent efficient, a certain amount of energy is lost in each stage and is thus reconverted into heat which is used in "reheating" the steam. This reheating of the steam in each stage thus makes a greater amount of energy available in the succeeding stages, since it increases the entropy and the total heat. Therefore the summation of the adiabatic energy drops of each stage is always greater than the adiabatic energy drop over the entire machine. Expressed in symbols, if

i_1 = initial total heat in the steam per lb., and if

i_2 = final total heat in the steam after adiabatic expansion from i_1 to the final back pressure, then

$i_1 - i_2$ = the available energy in B.t.u. per lb. of steam; and if

Δi = the fraction of the heat drop available for a single stage, then, on account of the reheating mentioned above,

$$\Sigma \Delta i = R(i_1 - i_2) \quad [1]$$

where R is the so-called "reheat factor." Let

W = the work done in the turbine in ft-lb. per lb. of steam and

$A = \frac{1}{778}$ or heat equivalent of 1 ft-lb.

Then the turbine efficiency will be

$$\eta_t = AW/(i_1 - i_2) \quad [2]$$

and we might define the average individual stage efficiency as

$$\eta_s = AW/\Sigma \Delta i \quad [3]$$

But from Equation [1]

$$\frac{AW}{\Sigma \Delta i} R = \frac{AW}{(i_1 - i_2)} \quad [4]$$

therefore,

$$\eta_t = R\eta_s \quad [5]$$

R is always greater than unity since the return of heat to the steam due to the losses in the machine always makes $\Sigma \Delta i$ greater than $i_1 - i_2$.

In general, for a given ratio of initial and final pressures the greater the proportion of the machine in the superheat region, the greater the value of R. (See Fig. 4.)

If, therefore, the stage efficiency η_s remains constant as the initial superheat increases, the turbine efficiency will follow some such line as *A*-2 in Fig. 1. The difference between the curve *A*-2 and the actual curve *A*-3 must then be due to an increase in the average stage efficiency.

From data available it would appear that there is practically no change in the individual turbine stage efficiency with increase in the average superheat of the steam in the stage. The change in the average stage efficiency with superheat must then be due to the

change in the stage efficiency with change in the moisture content of the stages in the moisture region. The method of analysis, therefore, to find the average correction to apply to the stage efficiency is to correct for the moisture content in the stage on the assumption that the expansion takes place in thermal equilibrium, and that the reduced efficiency with increased moisture content is due to the mechanical effect of the moisture in the steam. No assumption of the probable degree of supersaturation of the steam during the expansion is thus made.

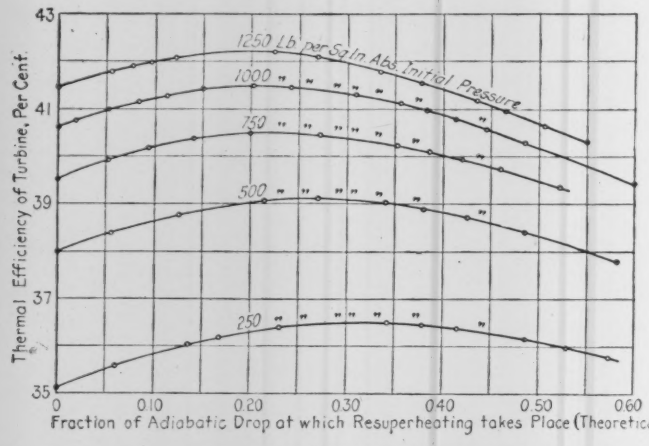

FIG. 2 THEORETICAL THERMAL EFFICIENCY OF HEAT CYCLES AT DIFFERENT INITIAL PRESSURES AND AT DIFFERENT POINTS OF RESUPERHEATING. (TURBINE EFFICIENCY 100 PER CENT)

Initial and resuperheating temperature 750 deg. fahr. Back pressure 1 in. Hg abs.)

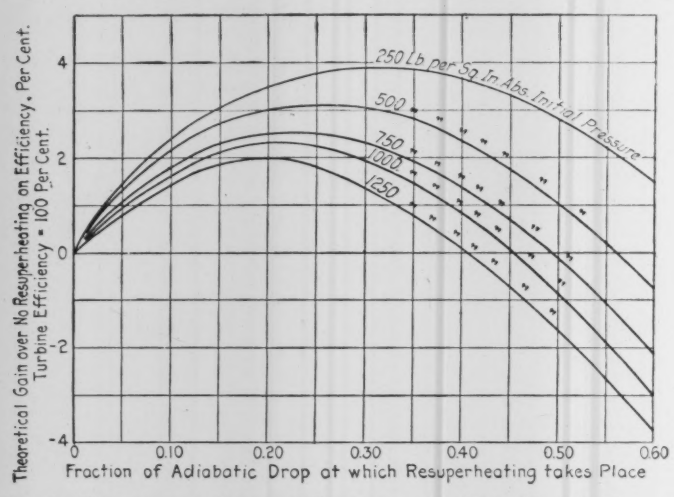

FIG. 3 THEORETICAL GAIN IN EFFICIENCY OF HEAT CYCLES DUE TO RESUPERHEATING AT DIFFERENT POINTS AND FOR DIFFERENT INITIAL PRESSURES

(Initial and resuperheating temperature 750 deg. fahr. Back pressure 1 in. Hg abs.)

DATA AVAILABLE AND ANALYSIS

Approximately twenty complete superheat-efficiency curves were available from ten different turbine combinations of the multi-stage impulse type; the initial pressures ranged from 60 to 200 lb. per sq. in. absolute, and the back pressures were all $1\frac{1}{2}$ in. Hg absolute. These curves have been obtained at constant speed, but at many different points on the speed curves of the different machines.

In the case where a turbine was designed for a resuperheating installation, a greater amount of bucket speed would be put into the machine to take care of the increased available energy and to keep the ratio of the bucket speed to jet speed constant. Therefore the superheat-efficiency curves had to be corrected to such a basis that the ratio of bucket speed to steam-jet speed was a constant. That is, it was assumed that the turbine was speeded up in proportion to the square root of the increased available energy as the superheat increased.

When these speed corrections had been made, the slopes of the curves with 60 lb. per sq. in. initial pressure and those with 200 lb. per sq. in. initial pressure were averaged separately. Then by a cut-and-try process an average stage efficiency and a correction for moisture were obtained which, if applied together with the correct reheat factor (which had been previously calculated), enabled calculated superheat-efficiency curves to be obtained. These calculated curves coincided with the average test curves to within a fraction of 1 per cent throughout the entire range of superheat from 0 to 300 deg. fahr. at both initial pressures. The correction for moisture came out in each case to be 1.15 per cent decrease in the turbine stage efficiency for each 1 per cent increase in the average moisture content in the stage.

APPLICATION TO RESUPERHEATING

The theoretical thermal efficiencies of the resuperheating cycles and the percentage gain in each case over the corresponding cycle without resuperheating are given in Figs. 2 and 3. It will be noted that the theoretical savings shown in Fig. 3 are comparatively small.

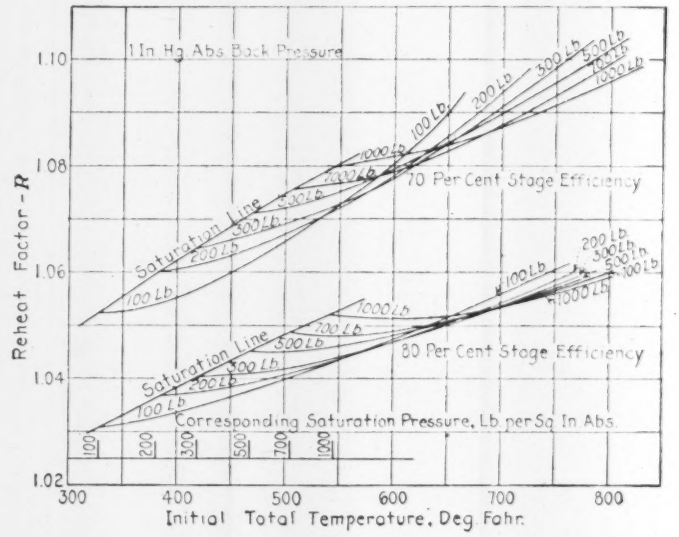

FIG. 4 REHEAT FACTORS FOR AN INFINITE NUMBER OF STAGES AT VARIOUS INITIAL PRESSURES, INITIAL TEMPERATURES, AND STAGE EFFICIENCIES

(Back pressure 1 in. Hg abs.)

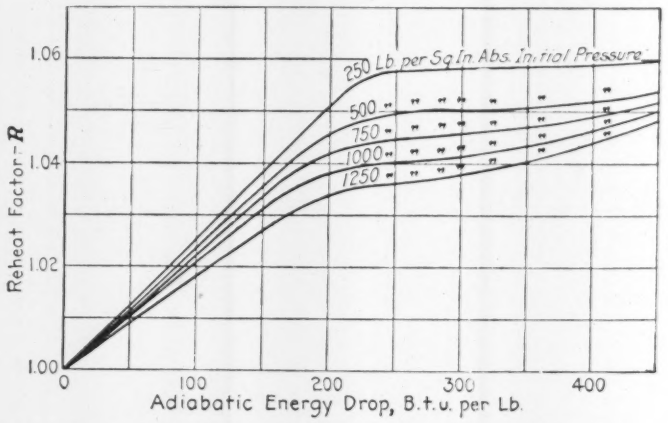

FIG. 5 REHEAT FACTORS FOR AN INFINITE NUMBER OF STAGES AT VARIOUS INITIAL PRESSURES AS FUNCTIONS OF THE ADIABATIC ENERGY DROP

(80 per cent stage efficiency, 750 deg. fahr. initial temperature.)

In general, the higher the pressure in a stage the lower will be its efficiency, due to the necessarily smaller nozzles and buckets and to the higher rotation and packing losses. Inasmuch as the stage efficiency is largely a matter of design, two assumptions were made regarding its variation with pressure: (1) That the dry-stage efficiency decreased in a certain manner as the pressure increased, and (2) that the dry-stage efficiency was constant throughout the turbine.

Due to the fact that the curves given in this paper do not include last-stage leaving losses, bearing losses, high- or low-pressure pack-

ing losses, or generator losses, all thermal-efficiency curves should be considered as being relative and not absolute in value.

The reheat factors given in Figs. 4 and 5 are for a turbine having an infinite number of stages. A turbine having from fifteen to twenty stages will have a reheat factor in which the part greater than unity will have a value approximately 0.9 of the value with an infinite number of stages.

The reheat-factor curves given in Fig. 5 are all on the basis of an 80 per cent stage efficiency. In order to obtain the reheat factor for efficiencies other than 80 per cent, the following approximation was used:

If $x = R' - 1$, where R' is the reheat factor for an infinite number of stages at 80 per cent stage efficiency, it was assumed that the part x was proportional to the difference between 100 per cent and the per cent stage efficiency, which is very nearly true. These assumptions then make the reheat factor used

$$R = 1 + 0.9(R' - 1.0)\left(\frac{100 - \eta_s}{20}\right)$$

where R is the reheat factor used and η_s is the average stage efficiency in per cent.

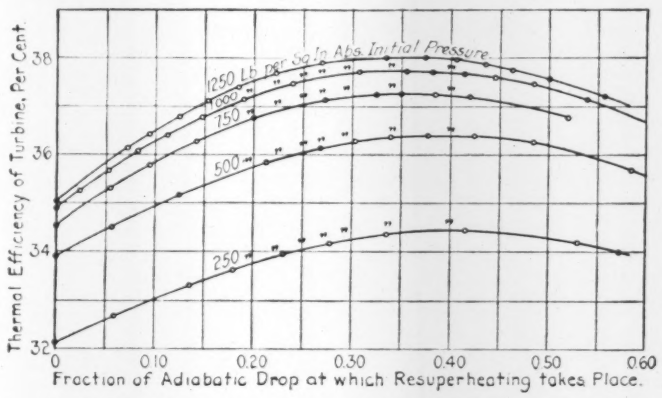

FIG. 6 THERMAL EFFICIENCY OF TURBINE FOR VARIOUS INITIAL PRESSURES AS FUNCTION OF POINT OF RESUPERHEATING

(Initial and resuperheating temperatures 750 deg. fahr. Back pressure 1 in. Hg abs. *Variable stage efficiency as a function of stage pressure.* No pressure drop in resuperheater.)

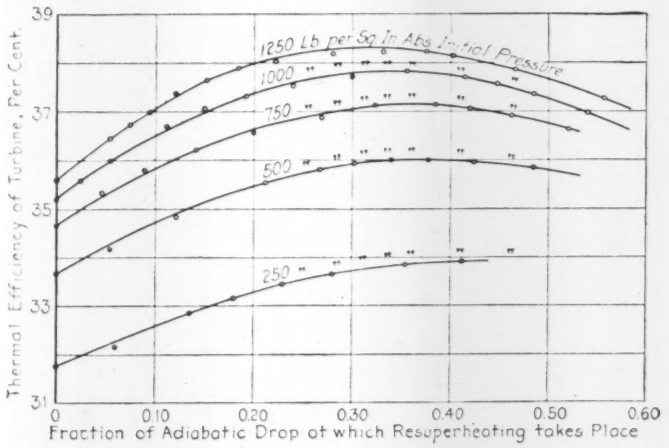

FIG. 7 THERMAL EFFICIENCY OF TURBINE FOR VARIOUS INITIAL PRESSURES AS FUNCTION OF POINT OF RESUPERHEATING

(Initial and resuperheating temperatures 750 deg. fahr. Back pressure 1 in. Hg abs. *Constant stage efficiencies.* No drop in pressure in resuperheater.)

RESULTS

Figs. 6 and 7 show the thermal efficiencies obtained for different points of resuperheating for the variable and constant dry-stage-efficiency assumptions and with 250, 500, 750, 1000 and 1250 lb. per sq. in. absolute initial pressure and an initial and resuperheat temperature of 750 deg. fahr.

Where the portion of the turbine under consideration was entirely in the superheat region, the end point D (See Fig. 8) was determined by multiplying the average stage efficiency by the reheat factor and subtracting the product of the result times the heat drop AC from the initial total heat A. Where a portion of the part of the turbine under consideration was in the moisture region, such as shown by the line EH, a "cut-and-try" process had to be used. An assumption was made regarding the location of point H and the per cent moisture at this condition was noted. It was assumed that the average moisture content per stage of the part of the machine in the wet region was one-half the final moisture content at H. Using the previous 1.15 per cent moisture correction and the assumed dry-stage efficiencies under consideration, the efficiency of that portion of the machine in the moisture region was calculated. On the assumption that the state curve from E to H was a straight line and weighing the efficiencies in the wet and dry portion of the machine according to the energy drops n and m in the wet and dry portions of the turbine, respectively, the average stage efficiency from E to H was determined. The turbine efficiency was obtained by multiplying this by the proper reheat factor obtained from Fig. 4 and the formula previously given for R. Knowing this value, the closeness of the first approximation to the location of H was checked.

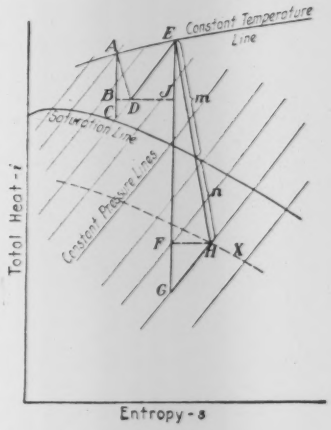

FIG. 8 TYPICAL EXPANSION LINES ON MOLLIER DIAGRAM

INCREASE IN THERMAL EFFICIENCY OF THE TURBINE WITH RESUPERHEATING

Figs. 9 and 10 were obtained from Figs. 6 and 7, respectively, by subtracting the thermal efficiency without resuperheating from the thermal efficiency with resuperheating at a given point and at a corresponding initial pressure, and dividing the difference by the thermal efficiency without resuperheating. These curves show the order of magnitude of the percentage gain in thermal efficiency due to resuperheating. No account has been taken of the pressure drop in the resuperheater and connected piping.

The gain, considering the change in turbine efficiency, is greater at the higher initial pressures, whereas the theoretical gain shown on Fig. 3, considering the heat cycle alone, is lower at the higher pressures. This is due to the fact that a greater portion of the turbine is operating in the moisture region with the high initial pressures and so is susceptible of greater improvement. The best point for resuperheating comes at approximately one-fifth to one-sixth of the initial pressure. This gives resuperheating pressures for initial pressures of 500 lb. per sq. in. and more, which do not require resuperheating piping of excessive size.

The greater gain shown by the assumption that the dry-stage efficiency decreased with increasing pressures in the stage is due to the fact that the gain in turbine efficiency from superheating is greatest in the low-pressure end where the efficiency is the highest under the variable-efficiency assumption. This results in a greater improvement than on the assumption of a constant stage efficiency.

To determine the effect of different initial and resuperheat temperatures upon the gain in thermal efficiency, calculations were made on the assumption that the initial and resuperheat temperatures were varied from 750 deg. fahr. (corresponding to the other calculations) to 550 deg. fahr. total temperature. The initial and resuperheat temperatures were assumed equal in all calculations, and the point of resuperheating was taken at a ratio of 0.355 of the adiabatic drop for the 750-deg.-fahr. condition with 1000 lb. per sq. in. absolute initial pressure. This put the resuperheating pressure at 150 lb. per sq. in., absolute. The curves of Fig. 11 show that the percentage gain in thermal efficiency is approximately constant at this resuperheating pressure irrespective of the maximum temperature of the cycle so long as the resuperheating temperature equals the initial temperature. This is in marked contrast to the theoretical gain, as also shown in Fig. 11.

Fig. 12 shows the percentage gain in thermal efficiency when reheating or resuperheating at various fractions of the adiabatic drop

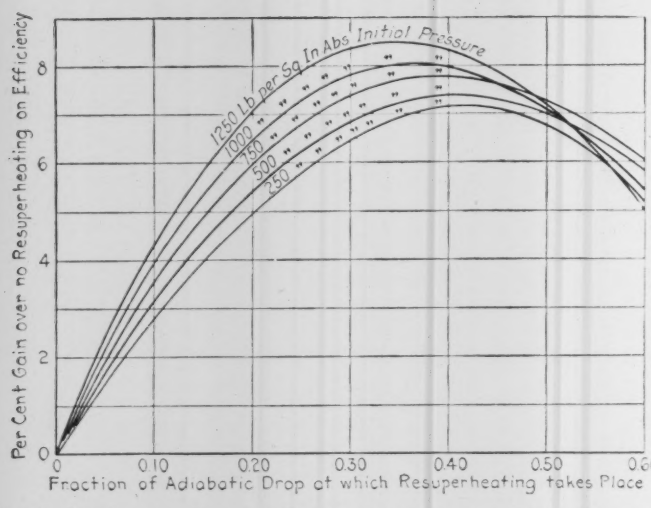

Fig. 9 Per Cent in Thermal Efficiency of Turbine Due to Resuperheating as a Function of Fraction of Adiabatic Drop after Which Resuperheating Takes Place

(Initial and resuperheating temperatures 750 deg. fahr. Back pressure 1 in. Hg abs. *Variable stage efficiency as function of stage pressure.* No drop in pressure in resuperheater.)

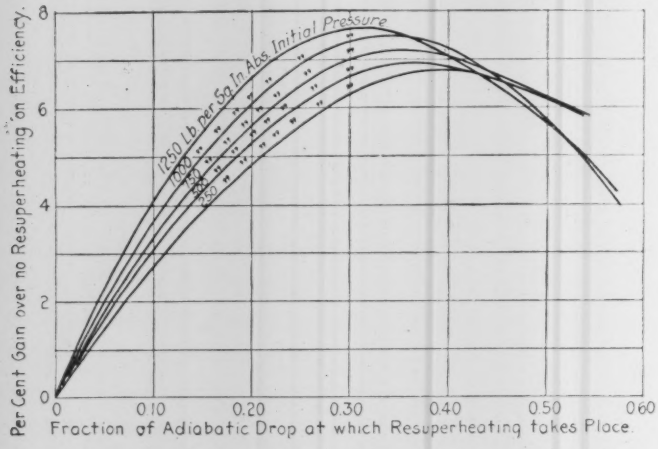

Fig. 10 Per Cent Gain in Thermal Efficiency of Turbine Due to Resuperheating as a Function of Fraction of Adiabatic Drop after Which Resuperheating Takes Place

(Initial and resuperheating temperatures 750 deg. fahr. Back pressure 1 in. Hg abs. *Constant stage efficiency.* No pressure drop in resuperheater.)

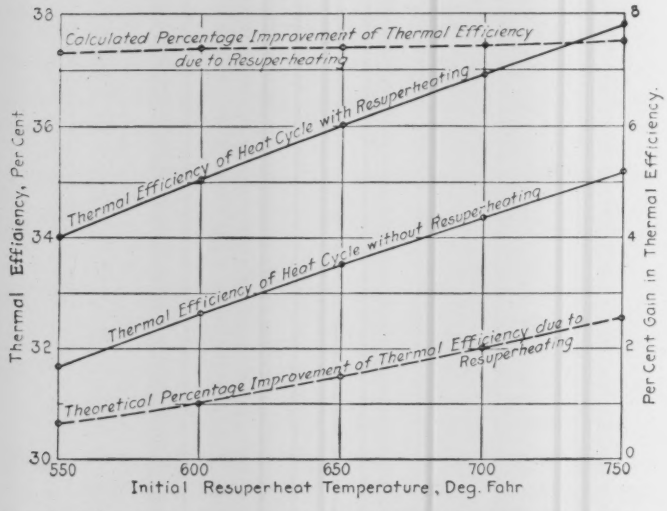

Fig. 11 Thermal Efficiency of Turbine and Per Cent Gain Due to Resuperheating as Function of Initial and Resuperheating Temperature

(Initial and resuperheat temperatures equal. 1000 lb. per sq. in. abs. initial pressure. Reheating at 150 lb. per sq. in. abs. Back pressure 1 in. Hg abs. No drop in pressure in resuperheater. Broken line at top of figure gives calculated percentage improvement of thermal efficiency of turbine due to resuperheating.)

with initial temperatures and resuperheat temperatures of 750 deg. fahr. in one case and in the other case equal to the temperature of saturated steam at the 1000 lb. per sq. in. absolute initial pressure. The fraction of the adiabatic drop at which it is best to resuperheat is only slightly changed by the initial and resuperheating temperature, and the maximum possible gain is somewhat greater in the case of the lower initial and resuperheating temperature. The best point for resuperheating is not necessarily the point at which the steam in the high-pressure turbine loses its superheat.

Increase in Thermal Efficiency of Turbine as a Result of Multiple Resuperheating

To reduce the number of laborious calculations a system was worked out whereby it was possible to obtain the gain due to multiple resuperheating from the results of the calculations for a single stage of resuperheating. The results of this calculation are shown in Fig. 13. If it is assumed that the drop in steam pressure in the resuperheaters and connected piping can be made so low as to reduce the available heat only one-half of one per cent for each resuperheating (this is somewhat lower than present practice), the net gains resulting from one or more resuperheatings will be as shown in Fig. 14. The gain in going from two to three stages of resuperheating is slight, but the increase in efficiency resulting from two resuperheatings as compared to one might warrant such an installation in the case of very large high-pressure base-load units.

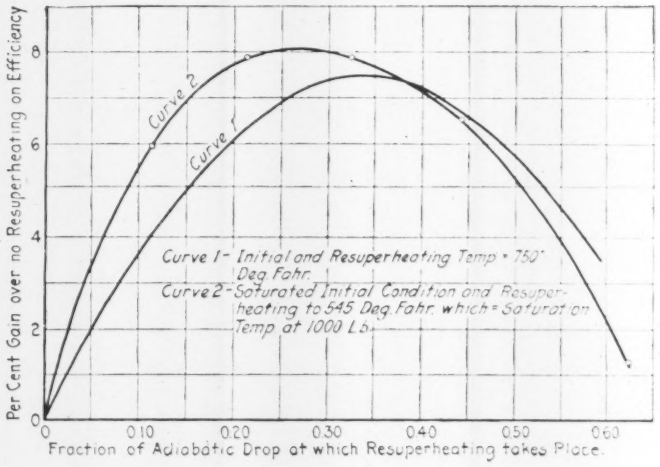

Fig. 12 Comparative Gain in Thermal Efficiency of Turbine Due to Resuperheating with Two Different Initial Temperatures at 1000 Lb. per Sq. In. Abs. Initial Pressure and 1 In. Hg Abs. Back Pressure

(No drop in pressure in resuperheater.)

Fig. 15 shows the relation between the initial pressure, fraction of the adiabatic drop at which resuperheating takes place, and the resuperheating pressure.

Effect of Resuperheating upon Exhaust Conditions and Turbine Capacities

Fig. 16 shows among other factors the variation of the total heat rejected to the condenser and the variation in the leaving loss of the turbine as the point of resuperheating is changed. Curves 2 and 6 show that per kw. of output the heat rejected to the condenser when resuperheating at 0.35 of the adiabatic drop is approximately 89 per cent of what it would be without resuperheating, so the condenser can be 11 per cent smaller for any given capacity.

Curves 4 and 5 of Fig. 16 show the reduction in leaving loss of a turbine of any given capacity with any given last-stage wheel as the point of resuperheating is changed. These curves show that the exhaust loss with resuperheating becomes 65 to 67 per cent of what it would be at the same capacity and for the same last-stage wheel but without resuperheating. This is due to the reduced steam volume in the last stage per unit of output, as shown on Curve 3 of Fig. 16, and on account of the higher heat drop per pound of working fluid.

The reduction in leaving loss has not been credited to the machine in the analysis of the gain due to resuperheating, since in

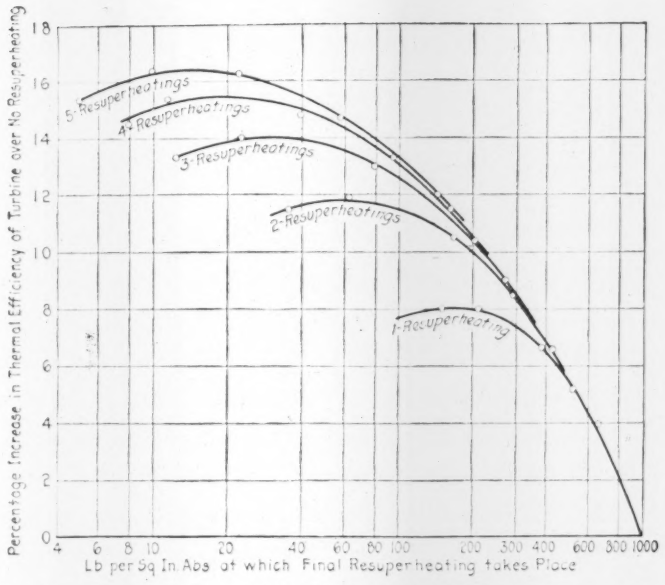

FIG. 13 COMPARATIVE GAIN IN THERMAL EFFICIENCY DUE TO USING ONE OR MORE STAGES OF RESUPERHEATING

(No drop in pressure in resuperheater. 1000 lb. per sq. in. abs. initial pressure. Each resuperheating to 750 deg. fahr. Final back pressure 1 in. Hg abs.)

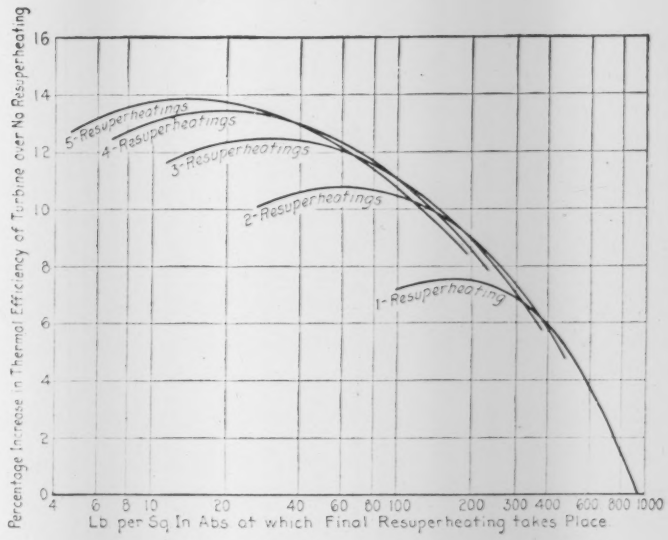

FIG. 14 PROBABLE ACTUAL COMPARATIVE GAIN IN THERMAL EFFICIENCY DUE TO USING ONE OR MORE STAGES OF RESUPERHEATING

(0.5 per cent reduction in economy assumed for each resuperheating due to pressure drop in resuperheater and connected piping. 1000 lb. per sq. in. abs. initial pressure. Each resuperheating to 750 deg. fahr. Final back pressure 1 in. Hg abs.)

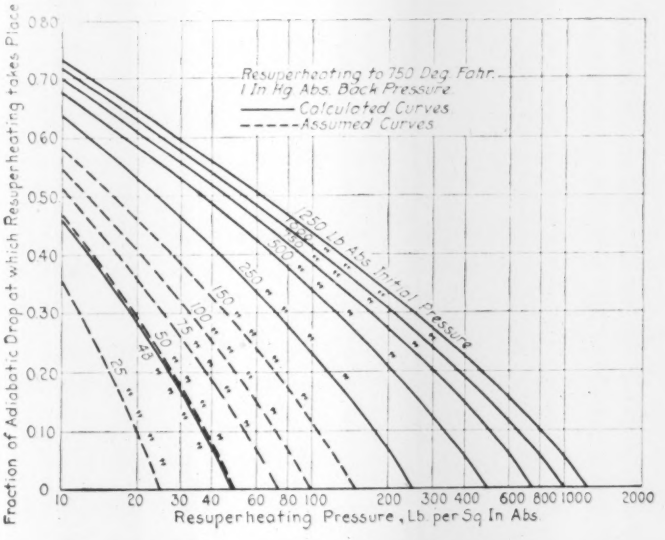
Resuperheating to 750 Deg. Fahr.
1 In. Hg. Abs. Back Pressure
Calculated Curves
Assumed Curves
250 Lb Abs. Initial Pressure
Fraction of Adiabatic Drop at which Resuperheating takes Place
Resuperheating Pressure, Lb. per Sq. In. Abs.

FIG. 15 RELATION BETWEEN PRESSURE OF RESUPERHEATING AND FRACTION OF ADIABATIC DROP AT WHICH RESUPERHEATING TAKES PLACE

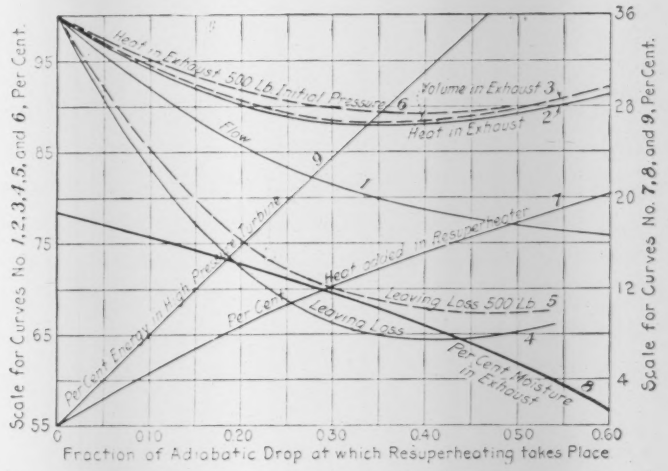
Heat in Exhaust 500 Lb Initial Pressure
Volume in Exhaust
Heat in Exhaust
Flow
Heat added in Resuperheater
Per Cent Energy in High Pressure Exhaust
Per Cent Leaving Loss
Leaving Loss 500 Lb
Per Cent Moisture in Exhaust
Scale for Curves No. 1, 2, 3, 4, 5, and 6, Per Cent.
Scale for Curves No. 7, 8, and 9, Per Cent.
Fraction of Adiabatic Drop at which Resuperheating takes Place

FIG. 16 VARIOUS FACTORS AS FUNCTION OF POINT AT WHICH RESUPERHEATING TAKES PLACE

(Unless otherwise noted on curve, initial pressure 1000 lb. per sq. in. abs.)

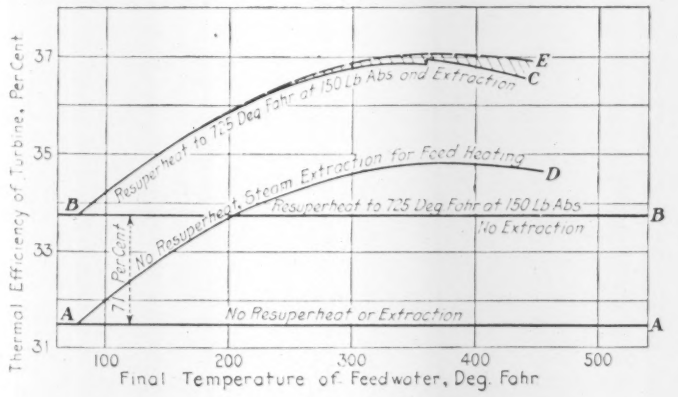
Resuperheat to 725 Deg Fahr at 150 Lb Abs and Extraction
Resuperheat, Steam Extraction for Feed Heating
Resuperheat to 725 Deg Fahr at 150 Lb Abs
No Extraction
No Resuperheat, Steam Extraction for Feed Heating
No Resuperheat or Extraction
Per Cent
Thermal Efficiency of Turbine, Per Cent
Final Temperature of Feedwater, Deg. Fahr

FIG. 17 CURVES SHOWING COMPARATIVE THERMAL EFFICIENCIES OF TURBINES WITHOUT RESUPERHEATING AND WITH STEAM EXTRACTION FOR FEED HEATING IN THREE STEPS AND OF TURBINES WITH BOTH RESUPERHEATING AND EXTRACTION

(Initial pressure 515 lb. per sq. in. abs. Resuperheating at 150 lb. per sq. in. abs. Initial and resuperheating temperatures 750 deg. fahr. Back pressure 1 in. Hg abs.)

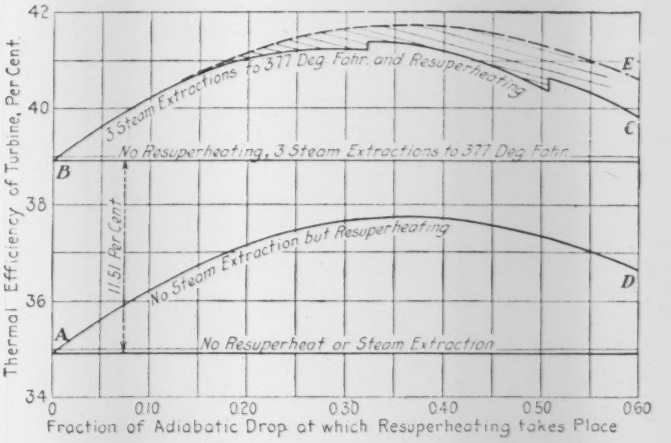
3 Steam Extractions to 377 Deg. Fahr. and Resuperheating
No Resuperheating, 3 Steam Extractions to 377 Deg. Fahr.
11.5% Per Cent
No Steam Extraction but Resuperheating
No Resuperheat or Steam Extraction
Thermal Efficiency of Turbine, Per Cent.
Fraction of Adiabatic Drop at which Resuperheating takes Place

FIG. 18 CURVES SHOWING COMPARATIVE THERMAL EFFICIENCIES OF TURBINES WITH RESUPERHEATING AND NO STEAM EXTRACTION AND TURBINES WITH BOTH RESUPERHEATING AND STEAM EXTRACTION IN THREE STEPS FOR HEATING FEEDWATER TO 377 DEG. FAHR.

(Initial pressure 1000 lb. per sq. in. abs. Initial and resuperheating temperatures 750 deg. fahr. Back pressure 1 in. Hg abs.)

most cases advantage is taken of this fact to increase the capacity of the turbine. Since for any given capacity the percentage leaving loss is only 0.65 of what it would be without resuperheating, the capacity of the turbine could accordingly be increased to $\frac{1}{\sqrt{0.65}}$ or 1.24 times what it would be with no resuperheating and with the same percentage leaving loss.

Gain Due to Combined Reheating and Steam-Extraction Regenerative cycles

It has also been thought of value, therefore, to investigate whether both the resuperheating and regenerative cycles could be used together, so as to obtain the sum of the two gains of the cycles when used separately. It has been found that under the conditions which would probably obtain, the total gain in thermal efficiency of the combination is equal to practically 95 per cent of the sum of the gains in efficiency when used separately. This is on account of the very small proportion of the heat in the extracted steam which is due to the superheat.

The curves of Figs. 17 and 18 have been calculated under somewhat different turbine-efficiency assumptions and steam conditions. The curves in each figure are, however, strictly comparable.

Fig. 17 shows a resuperheating and a non-resuperheating cycle with steam extracted at three equally spaced points for heating the feedwater in both cases. The curves were made so as to show the gain in each case when heating the feedwater to different temperatures. The feed-heating calculations were made on an ideal basis assuming no temperature drop, no pressure drop and contra-flow conditions in the heaters so that the drip from each heater left the heater at the temperature of the incoming feedwater. The drip was drained into the next lower heater in each case.

Curve *A-D* shows the thermal efficiency without resuperheating but with steam extraction for feed heating. Curve *B-C* shows the thermal efficiency with both resuperheating and steam extraction. Curve *B-E* is Curve *A-D* translated up a distance *A-B*, thus showing that the two gains practically add, one on top of the other. The cross-hatched area is the amount by which the total increase in efficiency, when using the combinations, fails to come up to the sum of the gains when used separately. The break in curve *B-C* occurs at the point where the upper heater reaches the resuperheating point. Curve *B-E* breaks away from curve *B-C* at approximately the point where the extracted steam becomes superheated.

Fig. 18 shows a similar set of calculations on the basis of resuperheating at various points in both a non-regenerative and regenerative cycle, *A-D*, being the non-regenerative cycle with resuperheating and *B-C* being the combined regenerating and resuperheating cycle. *B-E* is curve *A-D* moved up a distance *A-B*. It can be seen that the gains are nearly additive. The breaks in curve *B-C* in Fig. 18 occur at the points where the resuperheating point coincides with the extraction points. These breaks show that, wherever possible, it is more economical to extract at a pressure just above or at the resuperheating pressure and before the steam has been resuperheated, rather than at a point just below the resuperheating point.

Discussion

W. W. JOHNSON[1] emphasized and confirmed many important points in the authors' paper.

Frank O. Ellenwood[2] wrote that it was somewhat unfortunate that the results were based on data available to the authors only and since resuperheating became of greater value as higher pressures were used, that the maximum pressure considered had been only 200 lb. per sq. in. abs., data for higher pressures certainly being now available.

Except for Fig. 14 the authors had chosen to draw their conclusions without consideration of the throttling losses through the reheaters and piping system, and without any consideration of the relative costs of these systems for the various reheating pressures. The title of the paper probably justified the elimination of the item of cost, but for the central-station manager the selection of the proper reheating pressure certainly involved the cost of the reheating system and the space it occupied, as well as the increased thermal efficiency of the plant at various loads, when throttling was considered.

In Fig. 14 curves were given to show the relative advantages of multiple resuperheating, when allowing one-half of one per cent reduction in economy for the drop of pressure in each reheating system. These curves should have been calculated with progressively larger percentage losses for the correspondingly lower pressures involved when more reheaters were considered, because the actual losses were bound to be greater with the lower pressures.

The authors stated that in the actual turbine the "reheating of the steam in each stage thus makes a greater amount of energy available in the succeeding stages, *since it increases the entropy*." One might conclude that the best way to obtain the maximum available energy from steam would be to throttle it at every opportunity as much as possible since throttling was always a sure way to increase its entropy!

Linn Helander[1] wrote that the authors' data indicated that the best reheating pressure for a non-regenerative cycle occurred after approximately one-third of the adiabatic heat drop when the initial pressure was 500 lb. per sq. in. and the initial and reheat temperatures were 750 deg. fahr. Fig. 18 of the paper also supported the opinion that higher reheating pressures could be used more profitably for a regenerative than for a non-regenerative cycle. This figure showed, when losses occurring in the reheater piping were ignored, that the best reheat pressure for a non-regenerative cycle occurred after 37 per cent of the adiabatic heat drop, while for a regenerative cycle the best reheat pressure was higher and occurred after 32 per cent of the adiabatic heat drop. Losses in the reheater piping would widen this difference.

Contrary to the opinion of the authors, Fig. 18 appeared to indicate that the gain due to the combined regeneration and reheating was appreciably less than the sum of the gains in thermal efficiency due to regeneration and reheating used separately. The authors pointed out that the gain due to the combined use of regeneration and reheating was five per cent less than the sum of the gains obtainable from their separate use. This could not be ignored, since it was equivalent to one per cent of the total station fuel consumption, or approximately to the gain usually creditable to the third-stage heater of a three-stage feedwater heating system.

B. N. Broido[2] wrote that it was generally accepted and proved by tests that the decrease in efficiency for 1 per cent of moisture was considerably larger, being somewhere between 1.5 and 2 per cent instead of the 1.15 given by the authors. It was probable that the factor found by the authors of the paper applied to laboratory tests where the blades of the turbine were smooth and of correct shape. When a turbine was worked with saturated steam the blades were subjected to erosion, and the efficiency loss due to friction became greater, and the decrease of efficiency, due to the presence of moisture, was more pronounced.

The authors based their calculations for the increase in efficiency on the reheating being carried out to the final temperature of the initial steam in order to obtain the highest gain by reheating, and from a thermodynamic standpoint, the highest efficiency was to be obtained when the heat was added at the highest pressure. The authors had found that the most efficient point for resuperheating was at the ratio of about 0.35 of the adiabatic drop for a final temperature of 750 deg. fahr. This was theoretically correct. However, it would be difficult in practice to resuperheat the steam to 750 deg. With a pressure of 500 lb. per sq. in. and a final temperature of 750 deg. fahr. the superheat was 283 deg. Considerable space was required to accommodate a superheater of sufficient area to get this temperature, even when it was in the ideal location near the furnace.

Mr. Broido believed that if dependence had to be placed on the gases of the boiler for reheating, or if it should be necessary to conduct the steam from the turbine back to the boiler and then again

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[2] Chief Engr., Industrial Dept., The Superheater Co., New York, N. Y. Mem. A.S.M.E.

to the turbine, the gain to be obtained by resuperheating would hardly justify the complications. A much simpler method was to resuperheat the steam by live steam. While the final temperature of the resuperheated steam would be much lower than that of the superheated live steam, the arrangement was so simple that from a practical standpoint it was probably the most desirable. With 500 lb. per sq. in. pressure and 750 deg. fahr. final temperature, the steam would expand down to 70 lb. per sq. in. before it lost its superheat. It could be reheated by superheated live steam to a temperature of about 550 deg. fahr. The reheater could be located near the turbine, so that no long piping was required. The regulation of temperature was ideal, as all that was necessary was to adjust a valve admitting live steam to the reheater. Gains from combined reheating and regenerative cycles were indeed cumulative. There were, however, some objections to applying the regenerative cycle to a large turbine. The amount of water fed into the boiler was usually a varying one, which involved a varying amount of steam to be bled at the different stages. Further, it disturbed to a certain extent the relation between the stages, and this was necessarily accompanied by a decrease of turbine efficiency. Most large power plants were provided with a so-called house turbine to furnish power for all auxiliaries. If the steam for heating the feedwater was bled from a number of stages of the house turbine while the reheating cycle was applied to the main turbine, the two above-mentioned cycles would be fully utilized without interfering with each other and without bleeding any steam from the main turbine.

To accomplish the regenerative cycle, either a number of closed heaters or a number of pumps were required, together with the necessary piping, which was quite complicated. A new arrangement had been suggested recently which seemed to have overcome the difficulties. With this arrangement neither multiple pumps nor multiple heaters were required. The water to be heated was raised to a pressure enough higher than the boiler pressure to take care of friction losses through the device. This could be done by a standard feedwater pump. The feed line was gradually reduced in cross-section to a small size and then gradually increased to the original size of the pipe, similar to the reduction and increase of area in a venturi meter. As in the venturi meter, the operation of the device depended upon the law of hydraulics that the sum of static head, pressure head, and velocity head at any point in a pipe line equaled the sum at any other point, disregarding losses.

The section must be sufficiently reduced so that the pressure was less than the steam pressure of the lowest stage of the turbine from which steam was to be bled. The steam from this stage was introduced at this point and heated the feedwater to the temperature corresponding to the pressure. At a point further along the gradually increasing section, where the pressure, although higher than at the first point, was lower than the steam pressure of the next higher stage of the turbine from which steam was to be bled, steam was introduced from this stage, raising the temperature of the water to the temperature corresponding to the steam pressure. Steam from as many stages of the turbine as desired was thus introduced at points along the increase in area. In each case the water pressure was less than the steam pressure, so that the steam from the turbine could enter and mix with the water. After the last point of introduction of steam, the area of the cross-section was still further increased so that velocity head was transformed back to pressure head sufficient for the water to enter the boiler. With this arrangement it was possible to heat the water by bleeding from many stages without undue complication. All that was necessary for each stage was a pipe connection provided with a check value between the heater and turbine, so that all advantages in connection with multiple-stage heating could be obtained in a comparatively simple manner.

The authors, in closing, wrote that reliable data on superheating curves at pressures higher than 200 lb. per sq. in. were not available so far as they knew. Their data has been obtained in Schenectady by using a special battery of electric superheaters in order to vary the superheat before the steam passed to the throttle of the turbine. They had obtained a few isolated points on commercial machines in power stations and these data seemed to substantiate those obtained in Schenectady and to justify the figure of 1.15 they had used in all calculations. Recent tests made in Schenectady on single wheel stages in which the steam conditions had been varied from 300 deg. fahr. initial superheat to 15 per cent initial moisture content also substantiated the assumptions upon which the calculations had been based.

Regarding the point raised by Mr. Broido, that the gain was apt to be 1.5 per cent for each 1 per cent reduction in moisture content when the blades were corroded, they had made a number of tests on machines at different superheats and with different conditions of blades, and the indication was that the loss in the turbine due to moisture was in the nature of internal friction in the steam itself rather than friction along the sides, and that this internal friction was independent of the other losses in the stage and only there by virtue of the properties of the fluid itself.

Regarding the cost of a resuperheating installation, a general statement giving the percentage excess cost of a resuperheating turbine over a unit of the same rating to operate between the same initial and exhaust conditions but without the resuperheating feature, could not be made, since too many factors entered into the question. In general, the resuperheating type of unit was inherently more expensive for several reasons. In order to utilize the additional energy properly, the equivalent of a larger number of stages was necessary, and the introduction of hot steam into a turbine at points widely separated made necessary the use of steel to a greater extent than in the ordinary type of unit. In case a resuperheating unit should trip out for some reason, even though the main throttle valve might close, calculations showed that there might be enough energy available in the steam in the turbine, in the resuperheater, and in the connected piping to accelerate the unit to a dangerous speed. For this reason, an intercepting valve should be furnished just ahead of the point where the resuperheated steam entered the turbine. This valve and the gear which operated it were of course not required on the non-resuperheating type of unit.

In regard to the difficulty of resuperheating steam to 750 deg. fahr. at low pressure, the authors had already supplied several turbines to use this resuperheating feature up to 725 deg. fahr.

They had not included the effect of throttling losses in the resuperheater and connecting pipes, because this was a factor that entered into the design of the power station, the boiler, and the resuperheater and not of the turbine. Furthermore the determination of the correction to be applied to the curves in the paper in order to allow for this drop was comparatively simple.

Mr. Helander and Professor Ellenwood were undoubtedly right in pointing out that if the pressure drop in the resuperheater and connected piping together with the investment were taken into account, it would be more economical to resuperheat at a somewhat higher pressure than the peak of the curves given in the paper. The authors were inclined to believe that the variation in this pressure drop and in the cost of resuperheating equipment as a function of the pressure at which resuperheating was done, was a more important factor in determining the point at which it was most economical to resuperheat than the amount of the drop or cost in itself.

In connection with the combined resuperheating and regenerative cycles, there were always two questions which applied when a regenerative cycle was to be used. First, to what temperature would the feedwater be heated? Second, how many heaters or stages would be employed to heat the water to the given temperature? The first depended primarily upon whether economizers or air preheaters or neither were used in the boiler installation, and also upon the method used for auxiliary drive. The second was primarily an economic question, since the gain due to adding an additional stage of heating when the final temperature was fixed was considerably smaller than the gain resulting from adding the preceding heater, and a point was soon reached at which the cost of additional stages of feed heating with any given final feedwater temperature did not justify the additional cost and complication involved. This situation applied in either a resuperheated or non-resuperheated installation with only slight modifications to meet the individual conditions.

Professor Ellenwood had questioned the statement regarding the increase in the entropy of the steam resulting from the increased energy made available in the succeeding stages by the reheating. This would be perhaps somewhat clearer if changed to read, "since the reheating increases both the entropy and the total heat in the stage."

Gas Turbines

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The paper here abstracted presents a statement of the brake thermal efficiencies that may be obtained from gas turbines of various types, and discusses the possibilities and limitations of these types of heat engines. Calculated performances, with tables of computed values, are given for the following types: (1) Explosion turbine with regeneration, (2) Explosion turbine with regeneration and air cooling, (3) Explosion turbine with regeneration and water injection, (4) Explosion turbine combined with steam turbine operated from exhaust-heat boiler, (5) Explosion turbine combined with steam turbine operated from exhaust-heat boiler, with steam turbine operated from exhaust-heat boiler, with charge precompression and reduced back pressure, (6) Constant-pressure-combustion turbine with regeneration, (7) Constant-pressure-combustion turbine with regeneration and with cooling of the buckets by steam jets, (8) Constant-pressure-combustion turbine with regeneration and with steam injection into the combustion space.

In the conclusion it is stated that a review of the possibilities of the gas turbine does not give much hope of realization of efficiencies such as would encourage attempts to overcome the many difficulties with which the gas turbine is surrounded.

In two appendices not included in the present abstract are considered the entropy chart for gases and the representation of the proposed cycles on this chart, as well as the fundamental formulas and methods used in the computation of the data included in the tables.

THE BRAKE thermal efficiencies that may be obtained from gas turbines of various types are functions of the efficiency of the compressor and of the efficiency ratio of the turbine (turbine efficiency). No attempt is here made to estimate the compressor and turbine efficiencies which may be realized now or in the future, but calculations of brake thermal efficiencies have been made for a range of compressor and turbine efficiencies which, it is thought, includes both the present and the future possibilities.

The only published values of the attainable efficiencies of gas turbines, calculated on the basis of the true (variable) specific heats of the gases,[3] are contained in the fifth edition of Stodola's Dampf- und Gas-Turbinen. These values are exclusively for turbines of the explosion type and for a certain specified range of conditions. The present paper includes also constant-pressure-combustion turbines. In addition there are given the efficiencies for the more promising modifications of the simple cycle.

Expectation of High Efficiencies not Justifiable

The common expectation of improved efficiency for a gas turbine as compared with a reciprocating engine rests on an analogy with the steam turbine. This expectation may be dismissed as baseless. For the same pressure and temperature range the non-condensing steam turbine is less efficient than the non-condensing reciprocating engine. The higher efficiencies obtained with steam turbines result primarily from the use of higher superheats and lower condenser pressures than are practicable with reciprocating engines. If the gas turbine were able to utilize a more extended temperature and pressure range than is possible with a reciprocating engine a similar result might be expected, but instead, as will be shown later, the gas turbine is subject to limitations of pressure and temperature which do not exist for the reciprocating engine, and consequently offers no possibility of increased efficiency. Furthermore, since the gas turbine operates with exhaust at atmospheric pressure, or, if an exhauster is used, with only moderate vacuum, the fluid-friction losses will be greater than in a steam turbine with high vacuum. The gas turbine will be found to have a lower brake thermal efficiency than a reciprocating engine operating throughout the same pressure and temperature range.

The Field of the Gas Turbine

For smaller sizes the gas turbine cannot compete with the reciprocating engine which is available in compact form, of moderate weight, and of considerably higher efficiency. For 1000 hp. or over the advantage in diminished size and weight might in certain cases cause its choice in preference to the more efficient Diesel engine. As compared with a steam plant, the absence of boiler and condenser equipment might in some cases be decisive, even though the brake thermal efficiency is not likely to exceed that of a first-class steam-turbine plant and the fuel is likely to be more costly.

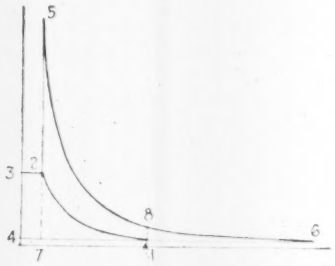

Fig. 1 Explosion-Turbine Cycle

Possible Gain in Efficiency from the More Complete Expansion of the Gas Turbine

The gas-turbine cycle has the advantage over the reciprocating engine of permitting complete expansion to the exhaust pressure.

In the *explosion-turbine* cycle, Fig. 1, the compressor work is 1234; the turbine work is 567 if the fresh charge is not admitted to the combustion chamber till after expansion is complete, but is 567 + 3274 if the fresh charge is admitted at the instant when the expansion has fallen to the pressure p_2. With expansion to the initial volume only, as in a reciprocating engine, the turbine work would be 7581 and 7581 + 3274 for the same two cases, respectively. The gain in cycle efficiency from complete expansion, V_6, compared with expansion to the original volume, V_1, is calculated in the complete paper for adiabatic compression, which is the condition approximated to in a reciprocating engine, and also for isothermal compression, which is the rational ideal in a multi-stage centrifugal compressor such as would presumably be used in a gas-turbine plant. The conditions assumed are: p_1 = 14.7 lb. per sq. in. abs.; t_1 = 200 deg. fahr. (366 deg. cent. abs.); exhaust pressure, 14.7 lb. per sq. in. abs.; gas fuel; excess air, 20 per cent; no heat losses; no utilization of heat of exhaust gases; compressor efficiency, 100 per cent; turbine efficiency ratio, 100 per cent. It is found that the gain in cycle efficiency is greatest with low ratios of compression. With a ratio of compression of 4 the increase in cycle efficiency with isothermal compression is from 27.8 to 39.3 per cent, or 11.5 per cent, which is an increase of 41.4 per cent. With adiabatic compression the gain is still greater, viz. from 18.5 to 34.6 per cent, or an increase of 87 per cent, but the actual efficiencies are lower.

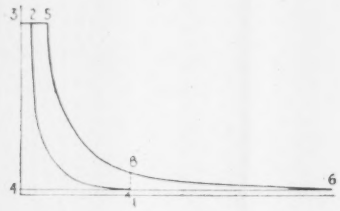

Fig. 2 Constant-Pressure-Combustion-Turbine Cycle

In the *constant-pressure-combustion* cycle, Fig. 2, the increase in work from complete expansion is shown by the area 861. The complete paper gives tables for gain in cycle efficiency for adiabatic compression and for isothermal compression. The conditions assumed are the same as for the explosion cycle except that the fuel is oil and the excess air is 100 per cent. The cycle efficiencies are naturally higher, owing largely to the much higher compression ratios proper to this cycle, and the gain from complete expansion is correspondingly less. For a ratio of compression of 10, with isothermal compression, the increase in efficiency is from 31.4 to 40.3 per cent, or an increase of 28.3 per cent; with adiabatic compression it is from 30.9 to 44.5 per cent, or an increase in efficiency of 44 per cent.

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[3] For efficiencies calculated on the basis of constant specific heat of the gases the reader is referred to the paper by Dr. H. N. Davis, "A *P-Q* Plane for Thermodynamic Cyclic Analysis," Proc. Am. Acad. of Arts and Sciences, vol. XL, pp. 629–654.

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Calculations in the paper show a high temperature at the end of expansion. The constant-pressure-combustion cycle is the more satisfactory in this respect with values from 928 to 821 deg. cent. abs. (1211 to 1018 deg. fahr.), which are not much beyond practicable temperatures of the present day. Reheating from turbine inefficiency would raise these temperatures very considerably in any actual turbine.

Practical Limitations of the Gas Turbine

Various practical difficulties result from the high temperatures involved. The turbine disk and buckets are necessarily subjected to high stress and, since they cannot be water-jacketed, a definite limit has to be set to the temperature to which they can be subjected. The temperature of the disk and buckets will be approximately the same as that of the exhaust gases, or, as it will be called here, the "casing temperature." In a simple gas turbine the casing temperature will be greater than the temperature of the gas discharged from the orifice as a result of reheating through bucket friction, windage, and residual kinetic energy. This reheating can be determined for any assumed turbine efficiency and in the cases considered in this paper is found to range from about 100 to 400 deg. fahr. By the use of special cooling devices, such as steam jets or cooling air, the casing temperature may be reduced even below the spouting temperature of the gases.

The present practical limit of casing temperature appears to be about 450 deg. cent. (850 deg. fahr.). Holzwarth is able to operate at this temperature with buckets of low-carbon iron. Rateau[1] reports a temperature of 1200 deg. fahr. in his exhaust-gas turbine with rotor of tungsten tool steel. As practically all steels have lost half their tensile strength at 1000 deg. fahr. there does not appear much chance at present of operating at temperatures in excess of that figure.

Selection of Type of Turbine

The limitation in casing temperature results immediately in another important limitation of the gas turbine, namely, the type of turbine which may be employed. Neither the reaction type nor pressure staging can be used, because even with the maximum possible pressure drop it is difficult to get the gases down to a temperature which is low enough to be practicable. With a single pressure stage, velocity staging is necessary in order to keep down the centrifugal stresses if the rotor is to run with a satisfactory peripheral speed. A practicable design can be made with two velocity stages and a corresponding bucket speed of about 22 per cent of the spouting velocity; the turbine efficiency estimated from steam-turbine experience will be less than 70 per cent, exclusive of windage losses. With three velocity stages and a corresponding bucket speed of about 15 per cent of the spouting velocity, the similar turbine efficiency will be less than 60 per cent. It appears, then, that a single-pressure-stage, two-velocity-stage impulse turbine is the only practical turbine arrangement at the present time; it has been accepted as such by all recent designers. The turbine arrangement being fixed, the maximum spouting velocity which can be efficiently utilized is determined by the maximum permissible peripheral speed, which, in turn, is limited by the strength of the rotor material at the casing temperature.

Limitation of Combustion Temperature

The limitations of casing temperature and of spouting velocity determine the maximum permissible temperature of combustion and thereby set definite limits to the attainable efficiency of a turbine operating without a special cooling device. The control of combustion temperature is through the control of the amount of fuel burned per unit weight of air compressed. With Diesel engines at full load it is usual to operate with 100 per cent excess air; with explosion engines the excess air is not usually greater than 30 per cent. It will be found that in a gas turbine, without special cooling device, the excess air must be increased very largely, with the unfortunate results that the ratio of the positive work done on the turbine, W_t, to the negative work of precompressing the air before combustion, W_c, is much less than in the usual reciprocating engine.

[1] *The Engineer*, Nov. 3, 1922.

Overall Efficiency

This ratio $R = \frac{W_t}{W_c}$ is most important in determining the efficiency of a gas turbine. The compression work has to be done in a separate machine, which would presumably always be a centrifugal compressor. The net work of the turbine is given by

$$W = W_t \times E_t - \frac{W_c}{E_c}$$

where E_t and E_c are the efficiencies of turbine and compressor, respectively, or

$$W = W_t\left(E_t - \frac{1}{RE_c}\right)$$

The value of E_t, as indicated above, will be less than 0.7; the efficiency of a centrifugal compressor E_c, for high ratios of compression, on the isothermal basis, is probably not greater than 0.7. Examining the factor $\left(E_t - \frac{1}{RE_c}\right)$, it will be seen that each of the three quantities in it should be as high as possible. With E_t and E_c both equal to 0.7, this factor becomes negative for $R = 2$; that is, no work can be obtained from the turbine under these conditions. Similarly for E_t and E_c both equal to 0.6, R must be greater than 2.77 in order that the turbine may be able to operate at all.

The quantity $\frac{W}{W_t - W_c}$ may be called the "overall efficiency" of the turbine; this may be written

$$\frac{W_t\left(E_t - \frac{1}{RE_c}\right)}{W_t - W_c} = \frac{W_t\left(E_t - \frac{1}{RE_c}\right)}{W_t\left(1 - \frac{1}{R}\right)} = \frac{E_tR - \frac{1}{E_c}}{R - 1}$$

Values for this quantity are given in the complete paper for varying compressor and turbine efficiencies.

Utilization of the Heat of the Exhaust Gases

Various procedures for utilizing the considerable heat carried away by the exhaust gases have been used or seem of sufficient promise to justify analysis:

1 The exhaust gases may pass through a regenerator for heating the fresh air. With a counterflow arrangement the entering air may be heated, after compression, to the temperature of the exhaust gases.

2 The exhaust gases may generate steam which:

- *a* May be sent into the combustion space in a constant-pressure turbine, or
- *b* May expand through steam nozzles and do work on the buckets (thereby cooling them) or on a special turbine on the same shaft, or
- *c* May drive a special condensing steam turbine and supply part or all of the power for operating the compressor, or
- *d* Various combinations of the above may be used.

Pressure Limits in Gas Turbines

There is no special condition limiting the compression pressure used in a gas turbine, but the increase in efficiency from high compression pressures is less than in reciprocating engines because the efficiency of centrifugal compressors falls off at high pressures in consequence of the high density of the air in the later stages and the resulting increase in frictional losses. The exhaust pressure will be in excess of atmospheric pressure when the heat of the exhaust gases is utilized in a regenerator or boiler, unless a gas exhauster is employed. As an exhauster acts on the gases after their volume has been diminished by cooling, its use may show an increase in thermal efficiency.

Many modifications of the gas turbine other than those mentioned above have been suggested and employed. The injection of water into the combustion space in conjunction with preheating the compressed air by the exhaust gases is analyzed in this paper. There have been several so-called gas-turbine projects employing an

oscillating water column, as in the Humphrey pump, and a hydraulic turbine. These are really water-piston engines and are not considered here. The most ingenious is that of Stauber.[1]

EXPLOSION TURBINES

Gas turbines may be classified in the same two groups as reciprocating engines: constant-volume-combustion or explosion turbines, and constant-pressure-combustion turbines.

In the explosion turbine the action is intermittent. The compressed air and fuel are introduced into the combustion chamber, the admission valve is closed, and the mixture is ignited and exploded; the nozzle valve is then opened and the products of explosion pass through the nozzle to the turbine buckets. After the pressure in the combustion chamber has fallen sufficiently, cooling air may be sent through the combustion-chamber nozzle and on to the turbine buckets, as in the Holzwarth turbine. The nozzle valve is then closed and the cycle begins again. Attention may be drawn here to certain features of this process. In order to operate, it is essential that the mixture in the combustion chamber should be explosive at the moment of ignition. This requirement limits the permissible amount of excess air, but to an extent which is not determinable from existing data on explosive mixtures. With air preheating, the temperature of the charge will be high—a condition favorable to ignition. If the fuel is injected very rapidly into air preheated to the ignition temperature and with the great turbulence which would exist under those conditions, the combustion may take place, as in a Diesel engine, with any quantity of fuel, however small, but the combustion though very rapid will not be a true explosion.

Another special condition which is inherent in the explosion turbine is that the spouting velocity of the gases during the expansion period will fall from a maximum at the instant of opening the nozzle valve to zero when the pressure in the combustion chamber is equal to that in the turbine casing. This is not a condition favorable to high turbine efficiency. Stodola[2] shows for a particular case, with initial spouting velocity of 4526 ft. per sec., that the velocity will have fallen to 3280 ft. per sec. when 66 per cent of the charge has passed, and to 1640 ft. per sec. after 96 per cent has passed. He finds a mean turbine efficiency of 63.5 per cent in this case as compared with an efficiency of 70 per cent for the most favorable gas velocity. These turbine efficiencies do not take into account windage losses.

CONSTANT-PRESSURE-COMBUSTION TURBINE

In the constant-pressure-combustion turbine there is a steady flow of air and fuel to the combustion chamber and a steady discharge of products of combustion through the nozzles. The temperature conditions offer much more difficulty than with the explosion type. Temperatures must be kept down either by using a very large excess of air, by injecting water or steam into the combustion space, by separate air or steam jets acting on the buckets, or by other device. This type, which has given highest efficiencies in reciprocating (Diesel) engines, does not promise similar efficiencies in the gas turbine in consequence of the low ratio of positive to negative work, R, and the low value of compressor efficiency, E_c, which results from the high compression pressures. On the other hand, turbine efficiencies, E_t, are higher than in the explosion turbine, as indicated in the preceding paragraph.

CONDITIONS FOR HIGH EFFICIENCY

The conditions assumed for the calculations are in some respects not practicable at present—this is especially true as regards possible casing temperatures.

No attempt is made to calculate probable turbine efficiencies, E_t, but it is evident that certain conditions, such as the reduction of windage losses, will favor higher efficiencies. The constant-pressure-combustion turbine, exerting a constant torque, will have a smaller percentage windage loss than the explosion turbine with intermittent applications of a lower average torque. A turbine with back pressure reduced by an exhauster will also have a smaller windage loss. On the other hand, an explosion turbine, like the Holzwarth, with short expansion period and prolonged cooling by low-pressure air, will have a low turbine efficiency especially if the turbine acts like an inefficient blower in speeding up the cooling as it passes through the buckets. Stodola[1] estimates the use of over 16 per cent of the turbine power to carry out the cooling by low-pressure air in this manner. With high-pressure air it is probably not less.

EFFICIENCIES OF VARIOUS CYCLES

The calculation of the efficiency of a gas turbine, using variable specific heats, can be carried out most readily by the use of a temperature-entropy chart such as is given by Stodola.[2] The process of calculation and the values of the specific heats used are given in an appendix to the complete paper. In most of the cases considered these calculations have been made for some assumed casing temperature and for a series of compressor and turbine efficiencies. The turbine efficiencies cover all the losses, including nozzle friction, bucket friction, windage, residual kinetic energy, and machine losses; that is, they are the ratio of brake work to the adiabatic total-heat drop from combustion-chamber conditions to exhaust pressure. All the energy lost in the turbine is assumed to be used in heating the gases at constant pressure from the temperature at the end of adiabatic expansion to the casing temperature. With a fixed casing temperature and a fixed combustion pressure this requires a different combustion temperature for each assumed turbine efficiency.

Certain conditions have been assumed for all the cases considered—with a few exceptions as noted:

a When gaseous fuel is used, it is assumed to have the volumetric composition: $H_2 = 11$; $CO = 22$; $CO_2 = 10$; $N_2 = 57$ per cent, and a low heating value of 123 B.t.u. per cu. ft. at 14.7 lb. per sq. in. abs. and 60 deg. fahr.

b When oil fuel is used, it is assumed to have the weight composition: $C = 86.5$; $H_2 = 11.2$; $N_2 = 2.3$ per cent, and a low heating value of 18,610 B.t.u. per lb.

c The effect on volume of molecular shrinkage due to combustion is neglected; the resulting error in the calculated efficiency is less than 1 per cent. The change in chemical compostion is in other respects taken into account as shown in Appendix No. 1 to the complete paper.

d The heat of combustion is assumed to be the same at constant pressure and at constant volume. The actual difference is less than 0.5 per cent. Similarly the variation of heat of combustion with temperature is neglected.

e The pressure drop between the air compressor and the combustion chamber is neglected.

f Compression efficiencies are all on the isothermal basis. A table in Appendix No. 3 to the complete paper gives the corresponding efficiencies on the adiabatic basis. The compressed air is supposed to have the room temperature when it reaches the regenerator or combustion chamber.

g The radiation loss to the walls of the combustion chamber is assumed to be 10 per cent of the heat of combustion. In most cases this heat is regarded as lost, but in certain specified cases it is utilized in generating steam.

h The pressure of the exhaust gases before passing through the regenerator or steam boiler is generally assumed to be 1.2 kg. per sq. cm. (17.06 lb. per sq. in. abs.) when exhausting to the atmosphere.

i Most of the calculations assume an initial air pressure of 1 kg per sq. cm. abs. (14.22 lb. per sq. in. abs.), which is 3 per cent less than the standard atmosphere. The difference between this pressure and the standard atmosphere may be considered as the pressure drop of the air entering the compressor. Its effect on efficiency is quite negligible.

j Spouting velocities are always theoretical quantities assuming no nozzle loss.

CALCULATED PERFORMANCE OF GAS TURBINES

A—EXPLOSION TURBINE WITH REGENERATION

The highest turbine-casing temperature which seems likely to be practicable, according to present indications, is about 500 deg

[1] Stodola, loc. cit., p. 1015.
[2] Loc. cit., p. 1007.

[1] Loc. cit., p. 1006.
[2] Ibid.

cent. (932 deg. fahr.). With exhaust gases at that temperature and with an efficient regenerator, a temperature of 450 deg. cent. (842 deg. fahr.) for the charge entering the combustion chamber should be obtainable. The efficiencies attainable under these conditions and with compression-pressure ratios 5, 10, 15 and 20 are given in Table 1. The temperatures after adiabatic expansion are the temperatures at which the gases would leave a frictionless expansion nozzle. The reheating due to nozzle and bucket friction, windage losses, and residual kinetic energy is the difference between the casing temperature and the temperatures after adiabatic expansion; in this case it ranges from 146 to 470 deg. fahr. The amount of excess air is so great (ranging from 533 to 1796 per cent) that no explosive mixture is possible; the cycle would presumably have to be carried out by injecting the fuel rapidly as in a semi-Diesel engine. The spouting velocities are the maximum values (disregarding nozzle friction) and diminish from the given values to zero. They are moderate and can be readily taken care of without necessitating excessive peripheral speed for the turbine buckets. The ratio of bucket speed to spouting velocity should be lower for gas turbines than for steam turbines because of their greater windage loss.

TABLE 1 CALCULATED PERFORMANCE OF EXPLOSION GAS TURBINES WITH REGENERATIVE HEATING OF THE COMPRESSED CHARGE TO 450 DEG. CENT. (842 DEG. FAHR.) AND TURBINE-CASING TEMPERATURE 500 DEG. CENT. (932 DEG. FAHR.)

Assumptions: Back pressure at gas turbine 1.2 kg. per sq. cm. abs. (17.06 lb. per sq. in. abs.); gas fuel; heat loss to walls of combustion chamber 10 per cent of heat of combustion; compression efficiencies on isothermal basis.

	Compression pressure, kg. per sq. cm. abs. (lb. per sq. in. abs.)																			
	5 (71.1)					10 (142.2)					15 (213.4)					20 (284.5)				
Turbine efficiency, per cent	55	60	65	70	75	55	60	65	70	75	55	60	65	70	75	55	60	65	70	75
Explosion pressure, kg. per sq. cm. abs.	6.80	7.00	7.17	7.35	7.54	15.05	15.55	16.10	16.70	17.30	23.6	24.6	25.6	26.6	27.7	32.7	34.1	35.6	37.2	38.8
Explosion pressure, lb. per sq. in. gage	82	85	88	90	93.5	197.5	200	214.5	223.5	231.5	321.5	335.5	349	364	369	450	470	492	515	538
Explosion temperature, deg. cent. abs.	988	1012	1038	1064	1091	1089	1125	1162	1207	1247	1145	1188	1235	1283	1338	1177	1232	1285	1343	1407
Explosion temperature, deg. fahr.	1319	1362	1409	1456	1504	1497	1566	1632	1713	1785	1602	1679	1764	1850	1947	1660	1758	1854	1958	2073
Temperature after adiabatic expansion, deg. cent. abs.	641	653	665	678	693	575	592	609	627	648	539	558	577	597	620	512	532	552	574	600
Temperature after adiabatic expansion, deg. fahr.	694	716	738	761	788	576	606	637	669	707	511	545	579	615	657	462	498	534	574	621
Excess air, per cent	1796	1622	1475	1342	1220	1235	1097	973	867	775	1032	908	800	706	622	935	815	708	615	533
Initial spouting velocity, ft. per sec.	2930	2980	3032	3085	3143	3535	3610	3690	3776	3862	3845	3942	4040	4143	4232	4068	4170	4280	4390	4490

Compressor efficiency, per cent	Brake thermal efficiency, per cent																			
55			2.3					0.7	8.46			2.6	10.5				4.2	12.6		
60			1.9	8.3				7.2	14.40			0.5	9.1	16.5			1.9	16.6	18.5	
65		0.6	7.3	13.3			5.50	12.8	19.50			6.5	14.6	21.5			7.9	16.1	22.3	
70		5.8	12.0	17.6		3.14	10.7	17.5	23.80			3.2	11.7	19.2	25.7		4.36	13.2	20.7	27.5
75	3.4	10.2	16.1	21.4		7.96	15.1	21.5	27.50			8.1	16.3	23.3	29.5		9.42	17.7	24.9	31.2

Blank spaces under brake thermal efficiencies indicate negative values, i. e.;—the turbine work is not sufficient to operate the compressor.

Stodola[1] gives charts showing the efficiencies of explosion turbines with regenerative heating of the compressed charge to 550 deg. cent. (1022 deg. fahr.) and with casing temperatures of 600, 800 and 1000 deg. cent. (1112, 1472 and 1832 deg. fahr.). Table 2 has been compiled from these charts. The tabulated values cannot be regarded as attainable unless there should be developed turbine-rotor materials capable of withstanding these high casing temperatures. The additional work required for cooling the rotor may be taken into account by assuming lower values of the compressor or turbine efficiency. With prolonged cooling by low-pressure air, alternating with the discharge of the hot explosion gases, the windage loss of the turbine will be much increased and the turbine efficiency correspondingly reduced. The higher the casing temperature (calculated on the assumption of no air cooling) the greater will be the negative cooling work.

TABLE 2 BRAKE THERMAL EFFICIENCIES, PER CENT, OF EXPLOSION GAS TURBINES WITH REGENERATOR

Data from Figs. 1048, 1049 and 1050 of Stodola's "Dampf- und Gas-Turbinen," 5th edition

Assumptions: Back pressure at gas turbine 1.2 kg. per sq. cm. abs. (17.06 lb. per sq. in. abs.); fuel, producer gas with low heating value 1100 kg. cal. per cu. m. (123 B.t.u. per cu. ft.) at 1 atm. pressure and 60 deg. fahr.; regenerative heating of compressed air to 550 deg. cent. (1022 deg. fahr.); heat loss to combustion chamber 10 per cent of heat of combustion; compression efficiencies on isothermal basis.

Compression pressure, kg. per sq. cm. abs.	Compressor efficiency, per cent	Turbine efficiency, per cent																							
		55					60					65					70					75				
		50	60	65	70	75	55	60	65	70	75	55	60	65	70	75	55	60	65	70	75	55	60	65	70	75
Casing temperature 600° C. (1112 deg. fahr.)																									
5		−19.0	−12.7	−6.0	0.0	4.2	−10.3	−3.6	2.2	7.5	11.0	−2.7	3.7	9.4	14.0	17.2	4.1	10.1	15.2	19.4	22.8	10.4	15.9	20.5	24.5	28.0
10		−14.5	−7.0	−1.0	4.9	9.1	−4.7	2.1	7.9	13.2	16.2	4.1	10.0	15.4	20.3	23.7	11.5	17.0	22.0	26.3	29.9	18.8	24.8	28.5	32.3	36.1
15		−13.7	−6.0	0.5	6.0	10.5	−3.6	3.4	9.0	14.4	18.2	5.5	11.6	17.0	21.9	25.2	13.5	19.0	23.7	28.2	31.7	20.8	26.0	30.5	34.2	37.5
20		−14.0	−6.3	0.5	6.0	10.5	−3.6	3.4	9.2	14.5	18.7	5.8	12.0	17.3	22.3	26.0	14.3	19.4	24.5	29.0	32.8	21.5	26.8	31.0	35.0	38.0
Casing temperature 800° C. (1472 deg. fahr.)																									
5		0.4	3.7	6.7	9.2	11.3	5.0	8.2	11.3	13.5	15.5	9.5	12.4	15.2	17.6	19.5	13.5	16.3	19.1	21.0	23.0	17.2	20.0	22.7	24.7	26.2
10		2.9	6.9	10.3	13.3	15.5	9.7	13.4	15.8	18.7	21.0	14.5	17.8	20.7	23.3	25.6	19.5	22.6	25.4	27.9	30.0	24.3	27.3	29.8	32.3	34.6
15		3.2	7.9	11.5	14.5	17.3	10.0	14.0	17.3	20.5	23.0	16.0	19.5	22.6	25.3	27.8	21.5	25.0	27.8	30.2	32.6	27.2	30.0	32.7	35.0	37.0
20		3.0	7.9	11.5	14.8	17.3	10.0	14.4	17.3	20.5	23.0	16.4	20.5	23.2	25.8	27.8	22.5	26.1	28.5	31.4	33.7	28.5	31.3	34.2	36.5	38.7
Casing temperature 1000° C. (1832 deg. fahr.)																									
5		7.0	9.1	11.1	12.6	14.2	10.5	12.5	14.5	16.0	17.5	14.0	16.0	17.8	19.2	20.5	17.2	19.2	20.7	22.2	23.8	20.4	22.3	23.7	25.1	26.2
10		10.5	13.1	15.1	17.3	18.7	15.0	17.3	19.5	21.5	22.8	19.3	21.6	23.7	25.4	26.7	23.5	25.4	27.4	29.0	30.5	27.3	29.3	31.0	32.7	33.8
15		11.4	14.2	16.4	18.5	20.5	16.3	19.0	21.2	23.4	25.0	21.0	23.5	25.7	27.6	29.2	25.4	27.8	29.8	31.7	33.0	29.7	32.0	33.8	35.4	36.7
20		11.0	14.2	16.5	19.0	20.8	16.3	19.5	21.6	24.0	25.6	21.5	24.2	26.4	28.5	30.2	26.3	28.8	30.9	32.9	34.5	30.8	33.7	35.1	37.0	38.3

B—Explosion Turbine with Regeneration and Cooling

The conditions of Table 1 have been selected as apparently those giving maximum efficiency for an explosion gas turbine with regenerator but without any special cooling device. If the gases reach a temperature in excess of 500 deg. cent. (932 deg. fahr.) in the turbine casing, it becomes necessary to add some device to keep down the temperature of the buckets and turbine disk. If the cooling device is a jet of high-pressure air expanding on to the buckets, it will consume power for compressing the air, and all of this power will not be recovered at the buckets. If it is low-pressure air the buckets will do work on it. The magnitude of the power required for this purpose depends upon the details of the particular scheme used.

C—Explosion Turbine with Regeneration and Water Injection

Another possible method of lowering the casing temperature is by injecting water into the combustion space. This case has been calculated for combustion with 100 per cent excess air, a casing temperature of 500 deg. cent. (932 deg. fahr.), regenerative heating of the compressed air to 450 deg. cent. (842 deg. fahr.), and a compression-pressure ratio of 15. For each assumed turbine efficiency there is a definite weight of water which must be injected to give the desired casing temperature; this weight is found by graphic methods and is given in Table 3. The explosion pressures and temperatures and the spouting velocity are seen to be moderate but the possible brake thermal efficiencies are not very promising—not more than 20 per cent could probably be realized.

D—Explosion Turbine Combined with Steam Turbine Operated from Exhaust-Heat Boiler

Regenerative heating of the compressed air increases the explosion

[1] Loc. cit., Figs. 1048, 1049 and 1050.

TABLE 3 CALCULATED PERFORMANCE OF EXPLOSION GAS TURBINES WITH REGENERATIVE HEATING OF THE COMPRESSED CHARGE TO 450 DEG. CENT. (842 DEG. FAHR.), WITH WATER INJECTION INTO THE COMBUSTION CHAMBER, AND WITH TURBINE CASING TEMPERATURE 500 DEG. CENT. (932 DEG. FAHR.)

Assumptions: Compression pressure 15 kg. per sq. cm. abs. (213.4 lb. per sq. in. abs.); back pressure at gas turbine 1.2 kg. per sq. cm. abs. (17.06 lb. per sq. in. abs.); fuel, oil; excess air 100 per cent.

Turbine efficiency, per cent	55	60	65	70	75
Ratio of weight of water injected to weight of air and fuel	0.415	0.387	0.358	0.328	0.297
Explosion pressure, kg. per sq. cm. abs.	23.4	24.3	25.3	26.4	27.6
Explosion pressure, lb. per sq. in., gage	318	331	345	358	378
Explosion temperature, deg. cent. abs.	1134	1174	1220	1270	1320
Explosion temperature, deg. fahr.	1582	1654	1737	1827	1935
Temperature after adiabatic expansion, deg. cent. abs.	543	561	580	600	620
Temperature after adiabatic expansion, deg. fahr.	518	550	585	621	657
Initial spouting velocity, ft. per sec.	4190	4270	4360	4460	4560
	Brake thermal efficiency, per cent				
Compressor efficiency, per cent 55	7.7	11.3	14.8	18.5	22.3
Compressor efficiency, per cent 60	9.8	13.4	17.0	20.7	24.5
Compressor efficiency, per cent 65	11.8	15.3	19.0	22.6	26.5
Compressor efficiency, per cent 70	13.5	17.0	20.6	24.2	28.0
Compressor efficiency, per cent 75	15.0	18.6	22.3	28.0	29.7

The gas turbines built by Holzwarth are of this kind, with the steam turbine driving the compressor and with all the gas-turbine work available. As the casing temperatures are high, air cooling is also employed. This case has been calculated by Stodola[1] for casing temperatures of 600, 800 and 1000 deg. cent. (1112, 1472 and 1852 deg. fahr.) and for a steam-plant efficiency of 16 per cent. Any excess of steam-turbine work over that required to drive the compressor is added to the gas-turbine work. The efficiencies attainable are given in Table 4 and are seen to be higher than the corresponding efficiencies in Table 2, especially for the lower compressor efficiencies. It should again be remembered that the additional negative work imposed by the air cooling is not here taken into account, unless lower turbine efficiencies are assumed for this purpose. When air cooling is employed the exhaust-gas temperature falls, so that the attainable steam-plant efficiencies would be lowered and the brake thermal efficiencies correspondingly reduced

TABLE 4 BRAKE THERMAL EFFICIENCIES, PER CENT, OF EXPLOSION GAS TURBINES IN WHICH THE HEAT OF THE EXHAUST GASES IS USED TO GENERATE STEAM AND DRIVE A STEAM TURBINE

Data from Figs. 1051, 1052 and 1053 of Stodola's "Dampf- und Gas-Turbinen" 5th edition

Assumptions: Back pressure at gas turbine 1.2 kg. per sq. cm. abs. (17.06 lb. per sq. in. abs.); fuel, producer gas with low heating value 1100 kg. cal. per cu. m. (123 B.t.u. per cu. ft.) at 1 atm. press. and 60 deg. fahr.; heat loss to combustion chamber 10 per cent of heat of combustion; efficiency of steam boiler 80 per cent, of steam turbine 20 per cent, of steam plant 16 per cent; compression efficiencies on isothermal basis.

Compression pressure, kg. per sq. cm. abs.	Turbine efficiency, per cent: 55					60					65					70					75				
Compressor efficiency, per cent	55	60	65	70	75	55	60	65	70	75	55	60	65	70	75	55	60	65	70	75	55	60	65	70	75
Casing temperature 600° C. (1112 deg. fahr.)																									
2	15.2	16.3	17.5	18.4	18.8	17.0	18.0	19.0	19.8	20.4	18.5	19.5	20.5	21.3	22.4	20.2	21.2	22.2	22.8	23.2	21.7	22.7	23.5	24.4	24.8
4	14.7	16.4	17.9	19.1	20.3	17.2	19.0	20.5	21.5	22.9	19.7	21.5	22.8	24.1	25.1	22.3	24.0	25.3	26.5	27.5	24.7	26.2	27.6	28.5	29.7
6	12.8	15.0	16.5	18.2	19.5	15.7	17.8	19.5	21.0	22.4	18.8	20.8	22.5	23.9	25.2	22.0	24.0	25.4	26.7	28.0	25.1	27.0	28.5	29.7	31.0
8	11.6	13.8	16.0	17.8	19.2	15.1	17.3	19.2	21.0	22.5	18.8	21.0	22.8	24.3	25.8	22.2	24.2	26.0	27.5	28.7	25.5	27.6	29.3	30.5	32.0
10	11.3	14.0	16.0	18.0	19.6	15.0	17.5	19.8	21.5	23.0	19.0	21.3	23.5	25.0	26.5	22.8	25.0	27.0	28.5	29.9	26.4	28.5	30.3	31.8	33.2
Casing temperature 800° C. (1472 deg. fahr.)																									
2	19.6	20.4	21.1	21.6	22.2	21.0	22.0	22.5	23.0	23.5	22.6	23.3	24.0	24.5	25.0	24.2	25.0	25.5	26.0	26.4	25.8	26.5	27.1	27.6	28.0
4	20.1	21.1	22.5	23.4	24.1	22.3	23.5	24.5	25.5	26.3	24.6	25.8	26.7	27.7	28.5	27.0	28.0	29.0	30.0	30.7	29.4	30.5	31.5	32.4	33.2
6	19.5	20.8	22.3	23.2	24.1	22.0	23.5	24.8	25.7	26.7	24.8	26.2	27.3	28.3	29.3	27.5	29.0	30.1	31.0	32.0	30.5	31.7	33.0	33.8	34.5
8	19.0	20.7	22.2	23.2	24.3	22.0	23.7	25.0	26.0	27.2	25.0	26.5	27.8	28.8	29.8	28.0	29.5	30.7	31.7	32.7	31.2	32.5	33.9	34.9	35.8
10	18.8	20.7	22.2	23.5	24.6	22.0	23.9	25.3	26.5	27.7	25.3	27.2	28.5	29.7	30.8	28.6	30.3	31.7	32.7	33.8	32.0	33.5	34.8	35.9	37.0
Casing temperature 1000° C. (1832 deg. fahr.)																									
2	22.0	22.5	23.0	23.6	24.0	23.5	24.0	24.5	25.0	25.7	25.2	25.7	26.3	26.7	27.2	26.8	27.3	27.9	28.2	28.7	28.5	29.0	29.5	30.0	30.2
4	23.2	24.0	25.0	25.7	26.5	25.6	26.5	27.3	28.0	28.5	27.8	28.8	29.5	30.2	30.8	30.1	31.0	31.7	32.5	33.0	32.4	33.3	33.9	34.6	35.2
6	22.8	24.0	25.1	26.0	26.8	25.5	26.8	27.8	28.7	29.5	28.2	29.5	30.4	31.2	32.0	30.8	31.8	32.8	33.6	34.5	33.5	34.5	35.4	36.2	37.0
8	23.0	24.5	25.5	26.5	27.5	25.8	27.2	28.2	29.2	29.9	28.5	29.8	31.0	32.0	32.7	31.5	32.6	33.6	34.6	35.4	34.3	35.4	36.4	37.3	38.2
10	23.1	24.5	25.6	27.0	27.8	26.1	27.5	28.6	29.7	30.7	29.3	30.5	31.7	32.8	33.7	32.5	33.7	34.8	35.7	36.5	35.3	36.5	37.6	38.5	39.5

TABLE 5 CALCULATED PERFORMANCE OF EXPLOSION GAS TURBINES WITHOUT REGENERATION BUT WITH EXHAUST HEAT USED TO GENERATE STEAM AND DRIVE A STEAM TURBINE, WITH CHARGE PRE-COMPRESSION AND EXHAUST TO ATMOSPHERE

Assumptions: Gas fuel; excess air 50 per cent; back pressure at turbine 3 in. of water above exhaust pressure; temperature of charge entering combustion chamber 288 deg. cent. abs. (59 deg. fahr); heat lost to walls of combustion chamber = 10 per cent of heat of combustion

Turbine efficiency, per cent	Casing temperature, deg. cent. abs.	Casing temperature, deg. fahr.	Brake thermal efficiency when steam turbine work = compressor work, per cent	Brake thermal efficiency, per cent, including steam-turbine work (steam-plant efficiency 15 per cent): Compressor efficiency, per cent 55	60	65	70	75	Steam-plant efficiency when steam-turbine work = compressor work, per cent: Compressor efficiency, per cent 55	65	75
Ratio of compression = 3											
50	1448	2147	21.9	23.2	23.9	24.6	25.1	25.6	13.2	11.2	9.6
55	1414	2086	24.1	22.9	25.6	24.3	24.8	25.3	13.6	11.5	10.0
60	1380	2024	26.4	26.9	27.7	28.2	28.8	29.4	14.1	11.9	10.2
65	1346	1964	28.6	28.8	29.6	30.2	30.7	31.1	14.6	12.4	10.7
70	1312	1902	30.8	30.7	31.5	32.1	32.6	33.2	15.1	12.8	11.1
Ratio of compression = 2											
50	1501	2242	18.6	23.6	24.1	24.5	24.8	25.1	7.9	6.7	5.8
55	1472	2190	20.4	25.3	25.8	26.2	26.5	26.8	8.1	6.9	5.9
60	1443	2138	22.3	26.9	27.3	27.7	28.1	28.4	8.3	7.0	6.1
65	1414	2086	24.1	28.4	28.9	29.3	29.6	29.9	8.6	7.3	6.3
70	1385	2034	26.0	30.0	30.4	30.8	31.2	31.5	8.8	7.5	6.5
No compression											
50	1598	2417	11.97			23.7					
55	1580	2384	13.17			24.7					
60	1562	2352	14.35			25.7					
65	1544	2320	15.55			26.7					
70	1526	2288	16.75			27.8					

and casing temperatures for a given amount of fuel burned, or, if the casing temperature is fixed, increases the amount of excess air necessary to keep down the temperatures. It consequently increases the negative work of the compressor and diminishes thereby the overall efficiency. A better method of utilizing the heat of the exhaust gases is to generate medium- or low-pressure steam with it and to use this steam in a steam turbine. With a high exhaust temperature a combined efficiency of boiler and economizer of 80 per cent may be assumed as a possibility, and a steam-turbine efficiency of 20 per cent. The steam-plant efficiency is then 16 per cent, and, in most cases, the steam-turbine work will be more than is required for driving the compressor.

below the values of Table 4. The plant becomes very complicated with air compressor, gas turbine, boiler, steam turbine, condenser, feed pump, etc.

The conditions of Table 4 demand a large excess of air. A further analysis of this cycle is given in Table 5, in which the excess air is kept at 50 per cent, which would presumably insure an explosive mixture. The casing temperature will then vary both with the amount of reheating (turbine efficiency) and with the compression ratio. This case is investigated for no compression and for compression-pressure ratios of 2 and 3. The casing tempera-

[1] Loc. cit., Figs. 1051, 1052 and 1053.

tures are very high, ranging from 1039 to 1325 deg. cent. (1902 to 2417 deg. fahr.), so that cooling, with its attendant losses, is necessary; no account of these losses is taken in Table 5.

Three different efficiencies are tabulated. Column 4 gives the brake thermal efficiency in the case where the compression work is done by the steam turbine and no excess work is available from that turbine. The last three columns show the steam-plant efficiencies that are necessary if the steam turbine develops exactly the power required by the compressor; as this ranges from 5.8 to 15.1 per cent it may be regarded as a possibility. Columns 5 to 9 give the brake thermal efficiency of combined gas and steam turbines when the increase if the other conditions are unchanged, and this may necessitate a speeding up of the rotor and an increase in windage losses from this cause. The additional work of increased expansion of the explosion gases is offset by the work required to be done on the exhauster. Since the exhauster acts on the exhaust gases after they have been cooled (desirably to atmospheric temperature), the negative work of the exhauster may be less than the increase in work done on the turbine, and an actual increase in brake thermal efficiency may be possible. Table 6, in which the performance of a turbine plant of this type is given, shows but little change in brake thermal efficiency as compared with Table 5, which is the

TABLE 6 CALCULATED PERFORMANCE OF EXPLOSION GAS TURBINES WITH CHARGE PRECOMPRESSION AND WITH AN EXHAUSTER FOR REDUCING THE BACK PRESSURE AT THE GAS TURBINE. THE HEAT OF THE EXHAUST GASES IS UTILIZED IN GENERATING STEAM TO DRIVE A STEAM TURBINE

Assumptions: Oil fuel; excess air 50 per cent; temperature of charge entering combustion chamber 288 deg. cent. abs. (59 deg. fahr.); back pressure at gas turbine 0.2 kg. per sq. cm. (2.84 lb. per sq. in.) greater than exhauster pressure; ratio of compression = ratio of exhaustion; temperature of water fed to boiler 15 deg. cent. (59 deg. fahr.); heat lost to walls of combustion chamber = 10 per cent of heat of combustion.

Turbine efficiency, per cent	Casing temperature		Brake thermal efficiency when steam-turbine work = compressor work, per cent	Brake thermal efficiency, per cent, including steam-turbine work (steam-plant efficiency, 15 per cent)					Steam plant efficiency if steam-turbine work = compressor + exhauster work, per cent		
	deg. cent. abs.	deg. fahr.		Compressor efficiency, per cent					Compressor efficiency, per cent		
				55	60	65	70	75	55	65	75
Ratio of compression = ratio of exhaustion = 1.5											
50	1496	2234	17.4	22.2	22.7	23.1	23.5	23.8	8.4	7.1	6.2
55	1472	2190	19.1	23.7	24.2	24.7	25.0	25.3	8.6	7.3	6.3
60	1447	2145	20.9	25.2	25.7	26.2	26.5	26.8	8.8	7.5	6.5
65	1422	2100	22.6	26.6	27.1	27.5	27.9	28.2	9.0	7.6	6.6
70	1395	2052	24.3	28.1	28.6	29.0	29.4	29.7	9.3	7.9	6.8
Ratio of compression = ratio of exhaustion = 2											
50	1435	2124	21.6	21.5	22.4	23.1	23.7	24.3	15.0	12.9	11.2
55	1402	2064	23.7	23.3	24.2	24.9	25.5	26.1	15.6	13.4	11.5
60	1370	2006	25.9	25.1	26.0	26.7	27.4	27.9	16.3	13.8	11.9
65	1335	1944	28.1	27.0	27.9	28.6	29.3	29.8	16.8	14.2	12.4
70	1305	1889	30.3	28.9	29.7	30.5	31.1	31.5	17.4	14.8	12.8
Ratio of compression = ratio of exhaustion = 3											
50	1360	1988	26.6	19.6	21.0	22.1	23.1	24.0	26.0	22.0	19.1
55	1317	1912	29.3	21.9	23.2	24.4	25.4	26.3	27.2	23.0	20.0
60	1275	1832	31.9	24.1	25.5	26.7	27.7	28.5	28.4	24.1	20.8
65	1235	1764	34.6	26.4	27.8	28.9	29.8	30.8	29.8	25.2	21.8
70	1195	1692	37.3	28.6	30.0	31.2	32.2	33.0	31.3	26.4	22.9
Ratio of compression = ratio of exhaustion = 1. (No compression or exhaustion)											
50	1600	2421	10.4			22.4					
55	1585	2396	11.5			23.3					
60	1570	2369	12.5			24.1					
65	1555	2342	13.5			25.0					
70	1540	2315	14.5			25.8					

TABLE 7 CALCULATED PERFORMANCE OF CONSTANT-PRESSURE-COMBUSTION GAS TURBINES WITH REGENERATIVE HEATING OF THE COMPRESSED CHARGE TO 450 DEG. CENT. (842 DEG. FAHR.) AND TURBINE-CASING TEMPERATURE 500 DEG. CENT. (932 DEG. FAHR.)

Assumptions: Back pressure at gas turbine 1.2 kg. per sq. cm. abs. (17.06 lb. per sq. in.); gas fuel; heat loss to walls of combustion chamber 10 per cent of heat of combustion; compression efficiencies on isothermal basis

Compression pressure, kg. per sq. cm. abs. (lb. per sq. in.)	5 (71.12)					10 (142.24)					15 (213.36)					20 (284.48)				
Turbine efficiency, per cent	60	65	70	75	80	60	65	70	75	80	60	65	70	75	80	60	65	70	75	80
Maximum combustion temperature (deg. cent. abs.)	944	962	982	1002	1022	1023	1054	1085	1117	1151	1073	1106	1142	1181	1223	1106	1146	1190	1236	1283
Maximum combustion temperature (deg. fahr.)	1240	1272	1308	1344	1380	1382	1438	1494	1551	1612	1472	1531	1596	1666	1742	1531	1603	1683	1765	1850
Temperature after adiabatic expansion (deg. cent. abs.)	651	604	678	692	706	503	612	631	651	672	561	580	601	624	649	536	559	583	608	634
Temperature after adiabatic expansion (deg. fahr.)	713	736	761	786	811	608	642	676	712	750	550	585	622	664	709	505	547	590	635	682
Excess air, per cent	1580	1442	1325	1222	1127	1113	994	890	799	717	940	830	732	646	572	830	724	634	556	485
Spouting velocity, ft. per sec.	2662	2702	2739	2770	2800	3237	3290	3340	3387	3438	3535	3595	3653	3715	3780	3726	3790	3870	3950	4032

Compressor efficiency, per cent	Brake thermal efficiency, per cent														
55			3.1	9.2			6.1	12.5		.4	8.0	15.3		1.4	9.92	18.0
60		2.8	9.0	14.7		5.5	12.2	18.1		7.1	14.2	21.8		8.0	15.9	23.5
65	1.7	8.2	14.1	19.4		4.0	11.1	17.4	22.9		5.5	12.8	19.3	25.5		5.5	13.6	20.9	28.1
70	.2	6.8	12.9	18.5	23.5	1.5	9.3	16.0	21.8	26.9	2.6	10.7	17.5	23.7	29.5	2.4	10.9	18.4	25.3	32.0
75	4.8	11.1	16.9	22.2	27.0	6.4	13.7	20.1	25.6	30.4	7.7	15.4	21.7	27.4	33.0	7.5	15.5	22.6	29.2	35.6

steam plant has an efficiency of 15 per cent. It may be noted that, under these conditions of operation, precompression of the charge is not particularly valuable; for example, if compressor efficiency is 70 per cent and turbine efficiency is 60 per cent, the brake thermal efficiency with a compression-pressure ratio of 3 is 28.8 per cent as compared with 25.7 per cent when no compression is used.

Supplementary data for this case are given in a table in the complete paper which show that the spouting velocities are high and that, in this respect, the condition with no precompression of the charge offers least difficulty. On the other hand, it is the condition with highest casing temperature and consequently of maximum cooling work losses.

E—Explosion Turbine Combined with Steam Turbine Operated from Exhaust-Heat Boiler with Charge Precompression and Reduced Back Pressure

By the addition of an exhauster to a gas turbine, the density of the medium in which the turbine rotates can be reduced and the windage losses consequently diminished. Spouting velocities will

generally similar case without exhauster; any advantage is to be looked for principally in reduction of windage loss. The complexity of the plant is even greater than in the preceding case in consequence of the addition of the exhauster. The complete paper contains supplementary data, for the conditions of Table 6, on temperatures after expansion and spouting velocities.

F—Constant-Pressure-Combustion Turbine with Regeneration

The general advantages of constant-pressure combustion have been touched on previously. If no cooling device is used the maximum permissible casing temperature may be taken as 500 deg. cent. (932 deg. fahr.) and the corresponding regenerative heating to 450 deg. cent. (842 deg. fahr.). These are the same as the conditions assumed for the explosion cycle of Table 1. The calculated performance of this cycle is given in Table 7 for the same compression ratios as in Table 1; the efficiencies obtained are not very different in the two cases and are higher for the explosion cycle except at low compression-pressure ratios. The casing temperature limits the maximum combustion temperature and thereby

limits the amount of fuel that may be burned. The excess air consequently is very high, ranging from 485 to 1580 per cent. This results in a high ratio of the negative work of compression to the positive work of the turbine and thereby results in a low brake thermal efficiency.

G—Constant-Pressure-Combustion Turbine with Regeneration and with Cooling of the Buckets by Steam Jets

The constant-pressure-combustion turbine without cooling (Table 7) is necessarily of low efficiency. Air cooling entails considerable losses. Another possibility in cooling is to generate steam by the exhaust gases and to expand this steam through nozzles on to the gas turbine, which thus serves both as a gas and a steam turbine. The steam will be used inefficiently in this case as it will be operating as a non-condensing turbine, but, as it discharges as wet steam at 212 deg. fahr., it will serve as an excellent cooling medium and will reduce the casing temperature.

Two cases of this cycle have been analyzed. In both of them oil fuel is used with 100 per cent excess air, a bucket peripheral velocity of 800 ft. per sec. is assumed, the back pressure at the turbine is taken as 1.2 kg. per sq. cm. abs. (17.06 lb. per sq. in. abs.) and the exhaust gases are supposed to be cooled to 100 deg. cent. (180 deg. fahr.) above the steam temperature in the exhaust-heat boiler and then to go through a regenerator. It is also assumed in both cases that the 10 per cent of the heat of combustion which goes to the combustion-chamber walls is used in evaporating water in the jackets around it. The compression pressure is 7.10 kg. per sq. cm. abs. (101 lb. per sq. in. abs.). The steam-nozzle velocity coefficient is taken as 0.95. The combustion temperature is 1601 deg. cent. abs. (2370 deg. fahr.).

I—In the first case it is assumed that steam is generated at such pressure that it has the same spouting velocity as the gases (3600 ft. per sec.) so that it can be efficiently utilized by the turbine. The gas and steam turbines are assumed to have 70 per cent efficiency, the compressor, 60 per cent. Computations on this basis yield the results tabulated below under I.

II—The steam pressure necessary to give the required spouting velocity in case I is very high (1070 lb. per sq. in. abs.). It seemed desirable to make computations also for the case in which the steam acts only on the second row of buckets and consequently has a spouting velocity of 1800 ft. per sec. It was assumed for this case that the steam-turbine efficiency was 70 per cent, gas-turbine efficiency 75 per cent, compressor efficiency 65 per cent. The computed results are tabulated below under II.

	I	II
Weight of steam generated per lb. of exhaust gas, lb.	0.214	0.314
Pressure of steam, kg. per sq. cm. abs.	75	3.2
Pressure of steam, lb. per sq. in. abs.	1070	45.5
Gas temperature after adiabatic expansion, deg. cent.	783	783
Gas temperature after adiabatic expansion, deg. fahr.	1441	1441
Gas temperature of gas alone, deg. cent.	952	925
Gas temperature of gas alone, deg. fahr.	1745	1697
Quality of steam after adiabatic expansion	0.751	0.946
Quality of steam after reheating in turbine	0.845	0.97
Casing temperature with combined gas and steam, deg. cent.	687	640
Casing temperature with combined gas and steam, deg. fahr.	1269	1184
Brake thermal efficiency, per cent	20.5	19.9

It will be seen that the cases calculated above yield casing temperatures of 687 and 640 deg. cent., respectively, which are too high to be practicable. With larger excess air this condition could be remedied but at the cost of reduced brake thermal efficiencies. These efficiencies are already quite low. This case does not appear to offer appreciably better prospects than case F, and it adds the complication of a waste-heat boiler.

H—Constant-Pressure-Combustion Turbine with Regeneration and with Steam Injection into the Combustion Space

Another method of using the steam generated in an exhaust-gas boiler is to discharge it directly into the combustion space. With this process the pressure at which the steam is generated does not have to be so high as in case G-I, where the steam acts independently on the turbine buckets. It is assumed to be generated at 10 per cent above the compression pressure. As no additional cooling is used, the casing temperature is limited to 500 deg. cent. (932 deg. fahr.). The exhaust gases are assumed cooled to 300 deg. cent. (572 deg. fahr.) in the boiler, the compressed air is heated by the regenerator to 200 deg. cent. (392 deg. fahr.), and the feedwater is heated, by the exhaust gases, to 50 deg. cent. (122 deg. fahr.). In addition to the heat from the exhaust gases, the 10 per cent of the heat of combustion which goes to the jacketed walls of the combustion chamber is assumed to be used in generating steam. The back pressure at the turbine is 1.2 kg. per sq. cm. abs. (17.06 lb. per sq. in. abs.).

This case was calculated for an assumed excess of air. This gives a definite combustion temperature and also, for each compression-pressure ratio, a definite temperature at the end of adiabatic expansion. With a casing temperature limited to 500 deg. cent. this fixes the amount of reheating in the turbine and therefore the turbine efficiency.

When calculated for 100 per cent excess air, the turbine efficiencies must be over 85 per cent if the casing temperature is to be kept down to 500 deg. cent. (932 deg. fahr.), even for very high compression pressures.

With 200 per cent excess air, the ratio of weight of steam to weight of products of combustion is 0.144 and the temperature of combustion before the injection of the steam is 1273 deg. cent. abs. (1832 deg. fahr.). The calculated efficiencies are given in Table 8.

TABLE 8 CALCULATED PERFORMANCE OF CONSTANT-PRESSURE-COMBUSTION TURBINE WITH STEAM INJECTION INTO THE COMBUSTION SPACE, WITH REGENERATION TO 200 DEG. CENT. (392 DEG. FAHR.) AND WITH TURBINE-CASING TEMPERATURE OF 500 DEG. CENT. (932 DEG. FAHR.)

Assumptions: Oil fuel; excess air, 200 per cent; steam pressure, 10 per cent greater than compression pressure; back pressure, 1.2 kg. per sq. cm. abs. (17.06 lb. per sq. in. abs.); steam generated by cooling exhaust gases to 300 deg. cent. (572 deg. fahr.) and by 10 per cent of heat of combustion going through combustion-chamber walls; feedwater heated to 50 deg. cent. (122 deg. fahr.) by exhaust gases.

Compression-pressure ratio		40	17.6	10
Combustion temperature after steam admixture	deg. cent. abs.	1128	1120	1116
	deg. fahr.	1571	1557	1550
Temperature after adiabatic expansion	deg. cent. abs.	460	569	661
	deg. fahr.	369	565	730
Turbine efficiency ratio, per cent		54.9	64.2	76.1
Spouting velocity, ft. per sec.		4210	3840	3500
		Brake thermal efficiency, per cent		
Compressor efficiency, per cent	55		5.8	13.8
	60		9.5	16.8
	65	3.3	12.6	19.2
	70	6.7	15.2	21.4
	75	9.7	17.5	23.2

An examination of this table shows that with turbine efficiencies of 54.9, 64.2 and 76.1 per cent, the necessary compression-pressure ratios, if the casing temperature is to be kept down to 500 deg. cent., are 40, 17.6 and 10, respectively. With a turbine efficiency of 76.1 per cent and compressor efficiency of 75 per cent (both values beyond existing possibilities) the brake thermal efficiency is only 23.2 per cent.

Conclusions

The advantages which the steam turbine has over the reciprocating engine are chiefly high rotative speed, absence of cylinder oil, large power per unit, low weight, low attendance cost, compactness, simplicity, small weight, and higher efficiency. The gas turbine also has the first three items as probable advantages but the other items cannot be definitely claimed, at any rate in so great a degree as in the steam turbine. A simple gas turbine without use of the exhaust heat would have too low an efficiency to be practical. If a regenerator is added but no provision for cooling is made (cases A and F), the operating conditions have to be such that the efficiencies obtainable are very low. If cooling is provided the weight and complication of the plant increase. With the explosion turbine (which appears to be the most practical in view of the lower compression pressures required and the possibilities of cooling) the use of valves in the combustion chamber introduces a type of complication from which the steam turbine is free and which may be expected to be the source of much trouble.

A review of the possibilities of the gas turbine does not give much hope for the realization of efficiencies such as would encourage attempts to overcome the many difficulties with which this machine is surrounded. Even with the increase in compressor and turbine efficiencies which may be expected to result from further developments it seems highly improbable that brake thermal efficiencies as high as 25 per cent could be obtained. The maximum that has been claimed up to the present time is about 17 per cent. This figure should be compared with brake thermal efficiencies of about 34 per cent obtained with Diesel engines, and about 38 per cent with the Still engine.

Mechanical Problems of Hydraulic-Turbine Design

As Illustrated by the 70,000-Hp. Unit Recently Put into Operation at the Plant of the Niagara Falls Power Company

By WILLIAM MONROE WHITE,[1] MILWAUKEE, WIS.

THE art of hydraulic-turbine design in America received its greatest impetus upon the putting into operation of the first large turbines at Niagara Falls. Previous to that time such men as Francis had laid an excellent foundation for an American type of development as contrasted to what had been previously done in Europe.

The part taken by American manufacturers in the further development and perfection of hydraulic-turbine apparatus can probably be best evidenced by the statement that during the last twenty-four years there has been expended under the author's direction alone possibly a half-million dollars solely for development, invention, experimentation, and test of apparatus forming a part of what today is termed "the hydraulic turbine and its equipment." Notwithstanding this, there is yet much to be done, although we have rather definitely outlined the constants fixing the water passages to, through, and from the turbine, and have fairly definitely fixed the types of auxiliary apparatus to provide for the convenient, economical, and safe operation of the turbine.

Assuming, then, the general characteristics of a given turbine to have been determined from well-developed formulas, it is proposed here to deal with the mechanical problems of hydraulic-turbine design, taking up in order problems connected with the flow of the water through the casing, the guide vanes, the runner, the draft tube, and then the mechanical transmission of power and its governing.

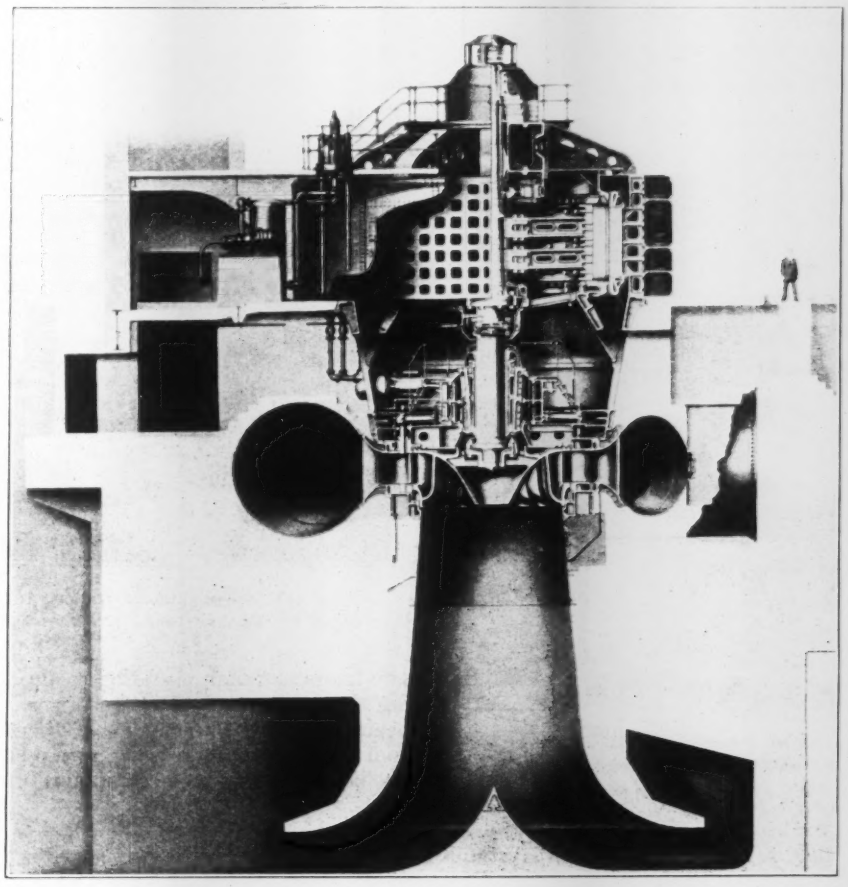

FIG. 1 COMBINED HYDROELECTRIC UNIT RATED AT 70,000 HP., 65,000 KVA. NIAGARA FALLS POWER CO.

(213 ft. head, 107 r.p.m. Plate-steel spiral casing, concrete hydraucone, direct-connected flyballs, adjustable lignum vitae bearing.)

We can probably best discuss the matter by taking a concrete example, and since we must have an example, let us take an outstanding unit such as the 70,000-hp. turbine recently put into operation at the plant of the Niagara Falls Power Company, Niagara Falls, New York. A cross-section through the turbine and generator of this unit, which operates under 213 ft. head at 107 r.p.m., is shown in Fig. 1. The first mechanical problem is to provide for the delivery of the water from the supply pipe to the runner. For this purpose a plate-steel circular-section spiral casing was used as illustrated in Fig. 2, and as shown in more detail with its speed ring in Fig. 3. In figuring the stresses for the casing one has to consider not only the normal operating head, but the maximum head due to increase of pressure in the casing caused by the inertia of the water when it is stopped by reason of the governor's rapidly closing the gates of the turbine for speed regulation. Provision must be made not for the static pressure alone but also for the shock resulting from this surge, which latter in some cases approaches water hammer. Thicker metal must also be used to provide for possible deterioration caused by rust and erosion on the interior and on the outside of the casing, and for this purpose $\frac{1}{8}$ in. has usually been added to the computed proper thickness of the plates.

THE CASING

Examination made of penstocks and plate work in service for twenty years and over shows that there is a surprisingly small amount of deterioration due to rust and erosion. Plate-steel spiral casings such as illustrated in Fig. 2 are usually imbedded in concrete, and it has been found advisable not to paint the outside of the casing as a firmer bond will thereby be secured between the casing and the concrete.

The greatest problem met with in the design of the plate-steel spiral casing for the 70,000-hp. unit at Niagara Falls was the driving of the $1\frac{3}{8}$-in. rivets, $6\frac{3}{4}$ in. long, through the flange of the speed ring and through the double thickness of plate, and also the driving of the $1\frac{3}{8}$-in. rivets at the lap joints of the larger section of the casing where the plates are $1\frac{1}{4}$ in. in thickness. For the purpose of driving these rivets a bull riveter was developed which was constructed by the Hanna Engineering Works in accordance with a model. This machine is capable of exerting 150 tons with 100 lb. air pressure and has a reach of 108 in. It is supported on a single cable and operates in any direction in order to accommodate itself to the various angles of the rivets about the speed ring and the lap joints of the casing. The riveter is adjusted to any required angle by means of two air motors, one of which rotates the jaws of the riveter and the other varies the angularity of the center line of the reach of the riveter. Fig. 4 shows this riveter in action. On actual test it drove sixty $1\frac{1}{4}$-in. rivets per hour, which is undoubtedly at a greater rate than could be accomplished on the usual fixed bull riveter. These rivets were of such size that it was thought inexpedient to attempt to drive them by hand. This riveter is now doing duty at the plate shops of the Allis-Chalmers Manufacturing Company's plant.

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The plates of the steel spiral casing in Fig. 2 are made of varying thickness from the large diameter around to the smaller section of the spiral, providing for a uniform stress of nearly 7000 lb. per sq. in., taking into consideration the efficiency of the joints. They are $1^1/_4$ in. thick at the large diameter, decreasing to $^5/_8$ in. at the small section. Flanged steel is used as material for most of the plates, and firebox steel for some of those requiring more difficult flanging. The interior of the plate-steel casing is not quite as smooth as that of the cast-steel, but its diameter is made larger, providing for sufficiently lower velocities to more than outweigh this difference. This can readily be done because of the lighter weight and lower cost of the plate-steel casing as compared with cast-steel casing.

A cast-steel speed ring illustrated in Fig. 3 serves to support the inner ends of the plates of the casing. This speed ring is made in five sections, all of cast steel, each section weighing 28,000 lb.

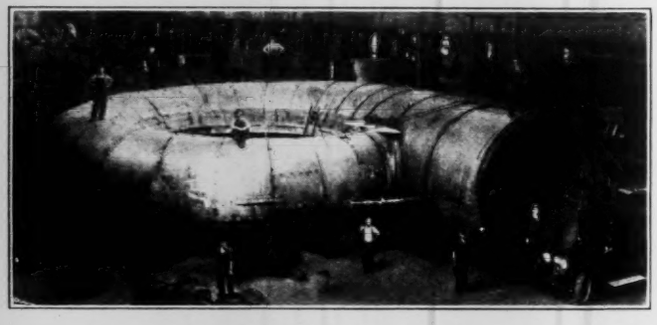

Fig. 2 Plate-Steel Spiral Casing for 70,000-Hp. Turbine

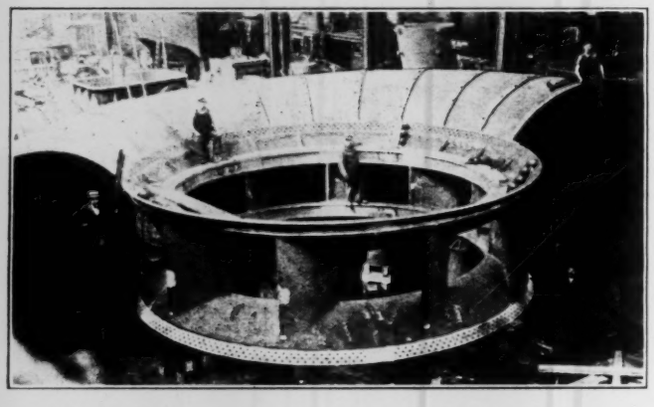

Fig. 3 Cast-Steel Speed Ring for 70,000-Hp. Turbine
(Made in five sections and held together with bolts and shrink links.)

It is no great accomplishment to compute the stresses in the steel casting, given the sizes and loads; the real mechanical problem lies in so shaping the design that it will have the least initial strains when cast. It will be noted in Fig. 3 that each of the five sections has two ribs or vanes. The upper and lower flanges are extremely heavy, but having only two vanes tying the two flanges together, any difference in the contraction of one vane on cooling will not produce stresses in the other vane; but should there be three vanes in each of the sections, the unequal cooling of any one of the three vanes would necessarily bring stress upon the others. In order to be assured of homogeneous metal at the junction of the vanes and the flanges or crowns of the speed ring, it is advisable to drill a pilot hole to the center of the mass at the junction of the vane and crown of the speed ring for examination. It will be noted in Fig. 3 that the sections of the speed ring are bolted together by heavy bolts; in addition a shrink link has also been found necessary to hold together the inner portion of each joint. A speed ring is a critical member in the design of any hydraulic turbine because the full flow of the water must pass between the ribs, which must be set at such an angle as to accommodate the flow with least disturbance and friction and yet be of sufficient thickness and shape to take the greatest strain imposed upon any part of the turbine. A wrong angle of the speed-ring ribs, or a wrong contour, may produce a disturbance in the flow which will extend through the guide vanes, through the runner, and into the draft tube. Any great disturbance in the flow at the speed-ring rib may tend to cause serious pitting of the runner.

Guide Vanes

The control of the water flow to the turbine introduces probably the greatest mechanical problem in its design. The almost universal method now in use is to place from 16 to 24 wicket vanes, or guide vanes, on a circumference just inside of the speed ring, and also to use an outside type of gate gear where the closed spiral is used. The diameter of the guide-vane bolt circle fixes the size and, consequently, the cost of the turbine, so that this dimension is made as small as is consistent with the proper guiding of the water to the runner with best efficiency. Experience has shown that a wedge-shaped guide vane provides for best efficiency. With the outside-type gear the vanes are supported by lugs or pivots cast at each side of the vane, with one lug extended for the connection of a lever for controlling the position of the vane. The stems on the guide vane must be large enough to resist the full water pressure against one side of the vane under full maximum-head conditions with a bearing pressure allowable for proper lubrication, and of a diameter such that the torsional stress is within

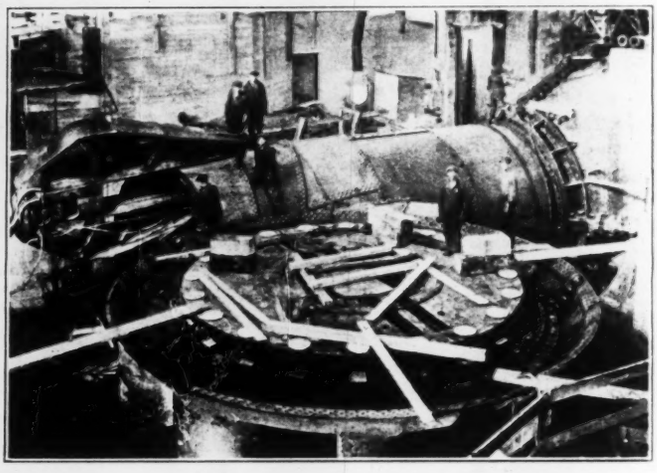

Fig. 4 150-Ton, 108-In. Reach, Adjustable Bull Riveter
(Capable of driving sixty $1^3/_4$-in. rivets per hour on spiral casing for 70,000-hp turbine.)

safe limits under all operating conditions. The bearings supporting the guide-vane stems are bronze-bushed, each lower bearing being lubricated through a hole drilled axially through the stem. Individual grease lubrication, now preferably of the Alemite type, has been found to be most suitable and convenient for the attendants.

The clearance between the edges of the guide vanes and edges of the adjoining distributor plates is fixed by the considerations of leakage on the one hand, and on the other by the necessity for sufficient clearance to minimize friction for governor operation. Too close clearances should not be attempted because the distributor plates at the ends of the vanes do rust as well as the ends of the guide vanes, decreasing the provided clearance.

Fig. 5 shows a partial section through a modern high-head Francis turbine for 350 ft. head. It will be noted that the stems of the guide vanes are extended and an outboard bearing provided near the end of the stem to take the forces produced by the actions of the operating lever, or guide-vane lever, secured to the end of the vane. A stuffing box should be provided at the exit of the stem from the cover to prevent leakage. This prevents the washing of the grease from the bearing as well as dirt from getting in, and although many attempts have been made to avoid this type of construction, it has been proven that it is not a useless luxury but is worth its additional cost. The guide vanes are all moved by the motion of one shifting ring, through suitable links, levers, and pins. The guide vanes are machined where each vane touches the next, and when new the leakage at the points of contactbetween the vanes is practically nothing because the clearance in a modern turbine does not exceed a few thousandths of an inch. The length

of the lever arms of the guide-vane stems and their relation to adjacent parts should be such that when the link is broken the outer end of the guide vane cannot swing into contact with the outer edge of the runner vanes. In the Allis-Chalmers Manufacturing Company's design the relation of the guide-vane lever with reference to the adjacent parts is such that the large end of the guide vane will not swing and touch the runner vanes.

For accuracy of adjustment of a vane with reference to an adjacent one, an eccentric pin is used either on the shifting ring or at the end of the guide-vane lever, whereby slight adjustments can be made. This adjustment is also useful in case the stem of a guide vane becomes sprung due to some foreign substance getting between any two guide vanes. A breaking point is now provided between the shifting ring and each guide vane whereby a part will be ruptured in case a foreign object is caught between two vanes. The shape of the guide vanes, the force of the governor, and the yield point of the breaking connection all have to be carefully and properly related to secure, on the one hand, instant relief under undue stress, and on the other hand freedom from too frequent and unnecessary interruptions.

Thrust bearings are provided on guide-vane bearings for thrust or weight in either direction axially of the vane, and adjustments are convenient for keeping the guide vane in position for equal clearances between the end of the vane and its adjacent distributor plate. In the case of vertical turbines for moderate and low heads, where the weight of the vane is greater than the upward thrust of the water pressure on the area of the guide-vane stem, then the upward-thrust bearing is omitted.

Cast-steel guide vanes are universally used for moderate- and high-head plants having outside gate mechanism, but for low-head plants having outside gate mechanism a plate-steel guide vane has been developed which is not only economical but even more suitable than the cast-steel because of the flexibility and malleability of the plate which permit it to be brought back to proper shape when distorted. Fig. 6 shows an 84-in. steel-plate guide vane under cross-bending test in a hydraulic press to determine ultimate strength and point of failure.

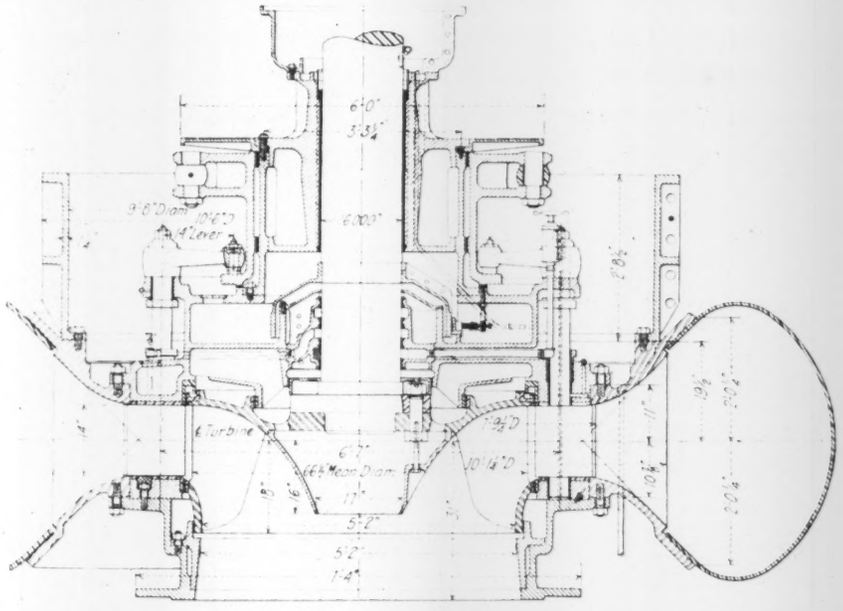

FIG. 5 SECTION OF 20,000-HP., 350-FT. HEAD, TURBINE, SHOWING OIL-LUBRICATED BEARING, VISCOSITY PUMP, GUIDE-VANE THRUST BEARINGS, AND RENEWABLE LINER BELOW RUNNER. SPEED, 360 R.P.M.

The velocity of the water as it discharges through the guide vanes is about as high as the velocity of flow through any part of the turbine. The cutting action of sand or other foreign substance in the water greatly increases with increased velocity, so that the parts adjacent to the ends of the guide vanes are usually faced with easily renewable plate-steel surfaces to provide for taking care of this wear without great expense. The clearance between the edges of the upper and lower crowns of the runner and their adjacent matching surfaces affects materially the efficiency of high-head units, and these clearances must be maintained at a minimum for maximum efficiency. For this reason it is advisable to have wearing rings around the periphery of the upper and lower crowns of the runner, and also renewable wearing rings in the adjoining places in the upper and lower crowns of the distributor. It is difficult to hold the wearing rings on the circumferences of the upper and lower crowns of the runner. The most satisfactory method is not only to shrink the rings on to the runner, but also to fasten them with screws as well. Frequently the rings are heated to such a temperature that they may be slipped over a shoulder and shrunk into a recess turned on the runner, which makes an excellent job but must be very carefully done. For medium and low heads where the water conditions are permissible these wearing rings are omitted.

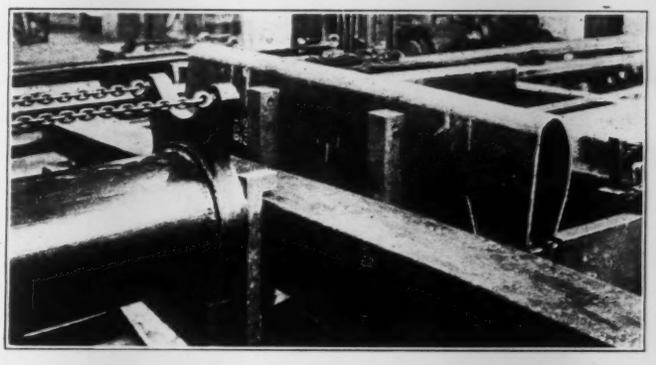

FIG. 6 84-IN. PLATE-STEEL GUIDE VANE BEING TESTED TO DESTRUCTION TO DETERMINE STRENGTH. USED ON MANY PLANTS UP TO 70 FT. HEAD

THE RUNNER AND ITS SHAFT AND BEARINGS

The turbine illustrated in Fig. 1 operates under a head of 215 ft. or a pressure of about 100 lb. per sq. in. This unit pressure on the large surfaces of such a turbine produces forces that are of astounding proportions. It is indeed a mechanical problem to cause proper flow through the runner and to provide runner vanes which will withstand the tremendous forces resulting under operating conditions. It is a simple matter to compute the forces acting on a vane of the runner due to the difference of pressure from one side of the vane to the other which would be caused by the velocity of the water under smooth streamline-flow conditions, but the water does not flow under such conditions and the factor by which to multiply the computed stresses for arriving at the probable stresses is one which is obtained only as a result of long experience. That region below the discharge edge of the runner vanes under abnormal conditions of head, and sometimes under normal conditions, may be subject to tremendous forces caused by eddies and vortices, and as these eddies and whirls break up and the vortices collapse, internal water-hammer conditions are caused which may rupture any cones or supports or mechanical plates placed in that region. The rumbling and low vibrations of turbines such as sometimes occur in normal practice arise from this action of the water. In some plants this may not be noticeable, but in others it is accentuated, because the natural period of some structure may correspond closely to that of the disturbance.

In order to provide for repairs to pitted runners a method has been worked out as illustrated in Fig. 7. This improvement consists in casting depressions in the vanes on the discharge side near the small opening to the next vane and setting therein steel plates held to the body of the vane by rivets, thus providing means for easy removal. Due to the almost mechanical action of the vortices at the discharge of the runner, it has been found advisable in many cases to install wearing plates on the draft tube near the discharge from the runner as illustrated in Fig. 5.

When a large hydraucone such as that shown in Fig. 1 is constructed of concrete, the form work is expensive and costly. A plate-steel hydraucone as illustrated in Fig. 8 has been used quite extensively and has been found economical and a great time saver in the construction of the substructure of the power house.

The enormous forces brought on the runner by reason of the collapsing of the vortices above mentioned produce strains in all directions. The runner, however, must be held rigidly in a central position because of the close clearance around its periphery, consequently not only a strong but a stiff shaft is required. A shaft of nominal diameter, that is, one computed to take care of the

FIG. 7 RENEWABLE PITTING PLATES AT DISCHARGE EDGE OF RUNNER (One of eight 45,000-hp. runners for Duke Price Power Co.)

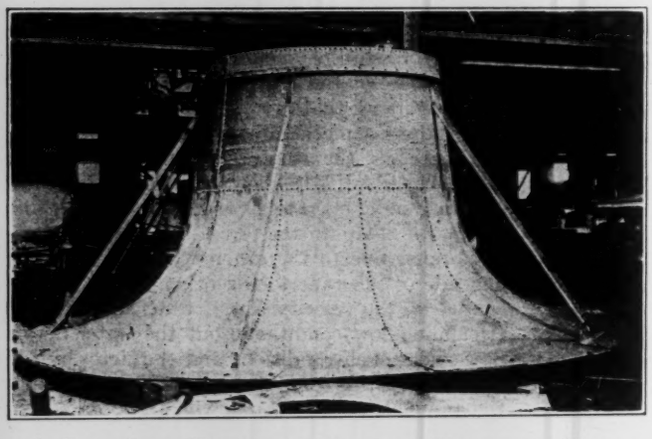

FIG. 8 PLATE-STEEL HYDRAUCONE ELIMINATES COMPLICATED FORMS AND SPEEDS UP CONSTRUCTION

torsional stresses only, could be made stiff by employing nickel steel, chrome-vanadium steel, or manganese steel, but stiffness can best be obtained by increase of diameter, consequently it is mechanically better to make the shaft of large diameter, using ordinary tough carbon steel instead of using a shaft of smaller diameter of the extra-quality material, either shaft costing about the same.

One of the important mechanical problems is the design of a proper guide bearing for the main shaft in the vicinity of the runner. Lignum vitae has been used for many years with considerable success. The tendency at the present time, however, is to the use of oil-lubricated, babbitted guide bearings. With this latter type of bearing satisfactory lubrication can only be accomplished by preventing, first, the oil from being drawn out of the reservoir by suction of the air along the shaft to the runner, and second, by preventing the water which leaks out of the same stuffing box, under conditions of high tailwater and high load, from getting into the oil reservoir supplying this bearing. The oil-lubricated, babbitted guide bearing is particularly advisable where the generator is so closely coupled to the water wheel that only one common guide bearing is used for the support of both the water-wheel runner and the water-wheel end of the shaft of the rotor of the generator. On 60-cycle generators the clearance between the poles and the yoke is rather small, and it is necessary to keep the clearance between the shaft and the bearing within close working limits.

The tendency of recent designs is to make each vertical guide bearing and its oiling system self-contained. Gear pumps are used in some cases to circulate the oil within the individual bearing. Another efficient device is the viscosity pump illustrated in Fig. 5 which provides for this lubrication. Still another arrangement is that shown in Fig. 9, where a flexible member such as a chain or spring passes around the shaft above the bearing and is led over pulleys at a right angle and down outside of the bearing into the oil reservoir below the bearing. Such a device was thoroughly tried out at the Green Island Plant of the Ford Motor Company.

The 70,000-hp. unit at Niagara Falls, although of the two-bearing type, has lignum vitae bearings, but the water in the Niagara River is so clear that serious wear does not occur and small clearances can be maintained. However, those bearings were made adjustable to provide for the maintaining of close clearances after the lignum vitae had worn away, and in addition a large settling

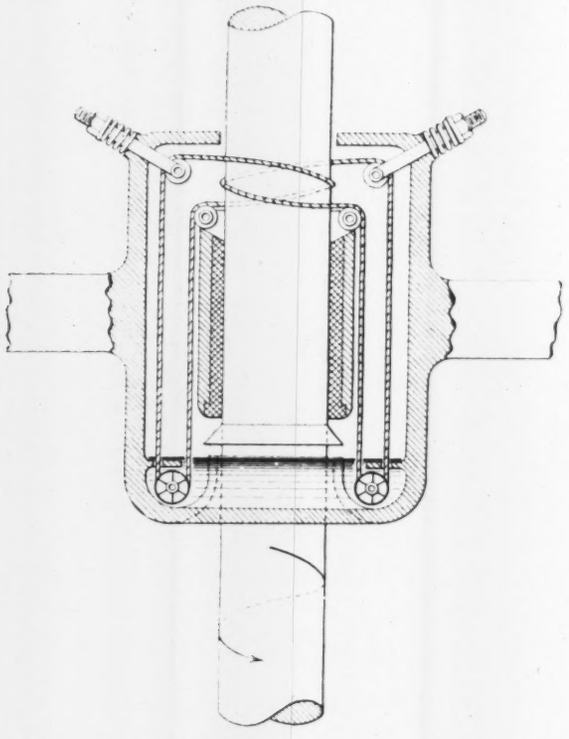

FIG. 9 FLEXIBLE SPRING DRIVEN BY SHAFT BRINGS CONTINUOUS SUPPLY OF FRESH OIL TO TOP OF BEARING. USED ON GREEN ISLAND TURBINES

basin was used at Niagara Falls which provided unusually clear water.

Practically all of the lignum vitae bearings built during recent years have been of the adjustable type. A very satisfactory bearing of this type consists of an outer housing and four or more cast bearing blocks designed to fit into the housing, the inner surface of the blocks being lined with strips of lignum vitae bored out to form the bearing surface. Bronze strips are provided where the blocks rest in the housing, and in case of wear shims may be inserted under these strips so as to bring the bearing surface out to its original position. By using shims of different thicknesses the blocks may be adjusted to any position to compensate for unequal wear.

Another type of adjustable bearing quite similar in construction may be adjusted by sliding the bearing blocks down farther into the housing guides, these being tapered. Still another type has the bearing blocks supported by eccentric or radial bolts, which provide for adjustment.

THE GOVERNING MECHANISM

The tremendous power of the modern hydroelectric units neces-

sitates a governing mechanism of great delicacy, because the slightest change of the gate mechanism means considerable change in the power output from the unit. The mechanical problem involved in securing this result is the providing of a positive and smooth drive for the flyball mechanism, since the perfection of governing is primarily dependent upon the fixed, continuous, and definite ratio of the speed of the flyballs with respect to the shaft and the speed of the unit. It has long seemed to the author that belts and gears have no place for the driving of the flyballs of large hydroelectric units, and to obviate the necessity of these a directly connected flyball has been developed which is mounted directly upon the main shaft. Such a device is illustrated in Fig. 1. This device has proven so delicate and so positive in its action that it has been incorporated in practically all of the hydroelectric units of the Allis-Chalmers Manufacturing Company's manufacture during the past six years. Delicacy of motion is secured by providing ball bearings at all moving points of the flyball. Positive motion of the valve is secured by using large flyball weights. This provides for such exact and positive gate motion that the gates are not constantly moved by a jiggling valve, but are only positioned when a change of load demands a new position of the gates. By using this direct-connected fly ball device wear and tear on the gate mechanism is reduced to a minimum. Oil has been found to be decidedly the best means for furnishing the fluid pressure for governor operation.

Recently a number of plants have been placed in operation where no governors are used, the gates of the unit being locked and only shifted when the load despatcher notifies the operator that the output of his station shall be changed. These units feed current into large systems where some of the units in other power houses are equipped with governors. Before the days of the large power systems, governors were a necessity, beause the load fluctuations frequently were a large percentage of the connected load. However, the percentage load fluctuations have not kept pace with the growth of transmission and power systems. A fluctuation of 1000 kw. on a 10,000-kw. unit was not considered extraordinary on an isolated system, but what 300,000-kw. system of today, except in case of abnormal trouble, receives an instantaneous load variation of 30,000 kw.? Probably an instantaneous change of 4000 or 5000 kw. represents the average fluctuation of the 300,000-kw. system of today, or a fluctuation of approximately one-sixth of that of the isolated system. The author feels that governing equipment will be simplified more and more, and that as a result many forced shutdowns now caused by failure of accessories and governing equipment will be eliminated, with an appreciable decrease in the cost of operation and an increase in the total output per kilowatt of installed capacity.

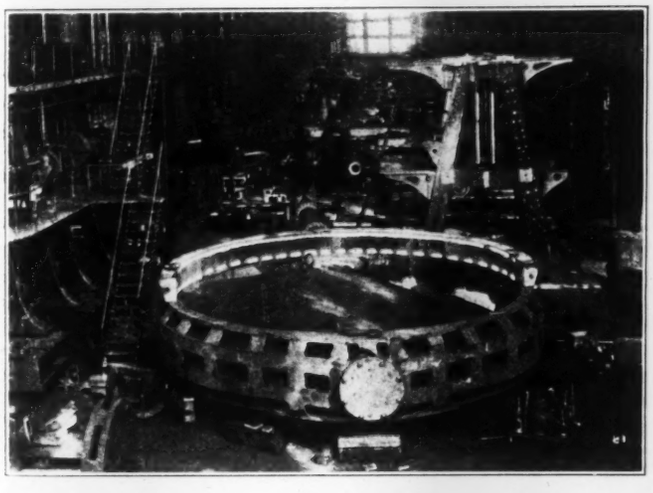

FIG. 10 LARGEST BUTTERFLY VALVE EVER BUILT BEING ASSEMBLED IN SHOP. DIAMETER 23 FT., 6 IN.; HEAD, 50 FT.

At the upper end of the penstocks feeding the three 70,000-hp. units at Niagara Falls are located three 23-ft. 6-in.-diameter butterfly valves, one of which is shown in Fig. 10. These valves take the place of the customary type of sliding head gates, and in addition are designed to open against full pressure or close with full velocity with a pressure of 50 ft. head on the center of the valve. Neither of these extreme operating conditions can be met by the ordinary type of head gate used in the majority of present-day plants.

The pivots on which the 23-ft. 6-in. wicket is carried is vertical, and when in closed position is designed to withstand a total force of 1,250,000 lb., this being divided on two 19-in. trunions. It being impossible to cast and ship this valve in one piece, the housing was made in six sections and the wicket in three sections, the wicket being 36 in. thick at the center section. The valve is operated by a hydraulic cylinder located 50 ft. above the valve and capable of moving the valve with 150 lb. oil pressure. This pressure is transmitted to the wicket through a 24-in.-diameter hollow shaft.

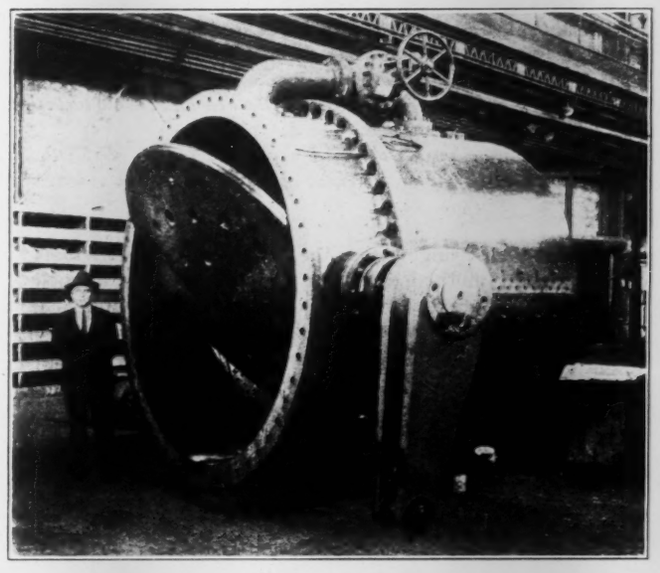

FIG. 11 96-IN. BUTTERFLY VALVE FOR 212 FT. HEAD. CAST-STEEL WICKET EQUIPPED WITH BRONZE PISTON RING DECREASES LEAKAGE

Butterfly valves may be made practically tight by inserting piston rings in the edge of the wicket. Such an arrangement is illustrated in Fig. 11, which shows a 96-in. valve designed to operate under 212 ft. head. Butterfly valves usually seat at an angle of from 8 to 12 deg. from normal for closed position. One half of the wicket is therefore wedged tighter by the water pressure but the other half is deflected, and unless some form of sealing device is used, some leakage will occur with large valves under high heads. By cutting a groove around the periphery of the disk, a bronze piston ring may be inserted which is pressed out by water pressure and springs so that the leakage will be insignificant; on a 108-in.-diameter valve under 421 ft. head it was less than one-fifteenth of a cubic foot per second.

On large vertical hydroelectric units the problem of obtaining a proper fit between the flanges of the turbine and generator shafts is difficult. Take a 25-ft. shaft 24 in. in diameter with 42-in. flanges. Suppose there is a discrepancy of 0.003 in. in one flange. This may cause the shaft to throw as much as 0.060 in., which is entirely too much. Yet it is no simple matter to machine such a flange within 0.003 in., and rather than risk a long delay when coupling up in the field, the Allis-Chalmers Manufacturing Company make it regular practice, whenever possible, to bolt the two shafts together in the shop and turn them in a lathe to test for trueness before shipping. Any unevennesses which could be quickly remedied in the shop would require days of fitting in the field, with still a doubtful job.

It has not been the author's intention to try to cover all the mechanical problems to be met in the design of a hydraulic turbine, as that is not possible. Only by continual experimenting, studying, and improving can we make hydraulic turbines of 100,000 hp. capacity operate uninterruptedly in order to serve continuously the many industries and homes that consume the output of these units and rely upon them for their living.

Investigations in Turning and Planing

By HANS KLOPSTOCK,[1] BERLIN, GERMANY

This paper gives an account of experiments conducted in the machine-tool laboratory of the Polytechnic Institute of Berlin. The cutting tests proper were preceded by a study of the lathe and the instruments used, to ascertain their behavior under various conditions. In the main tests it was aimed to determine the influence upon the cutting forces of the following factors: Cutting speed, chip section, shape of tool, and characteristics of the materials cut.

As a result of these observations a new form of tool has been designed which, the author claims, permits a higher cutting speed and a larger chip section than is practicable with the standard tools now used. The results of some of the tests made in various German railway shops are given, in which the relative life of the cutting edges of the new and the old tools under exactly the same conditions was compared.

It is claimed, from results obtained in various foreign shops, that on the average production can be increased about 30 per cent by means of the new tool.

WITH increasing wages, the problem of efficient tools and working methods becomes of ever-increasing importance. Scientific investigations covering turning and planing, which the author has carried on during a period of two years in the machine-tool laboratory of the Polytechnic Institute of Berlin, have

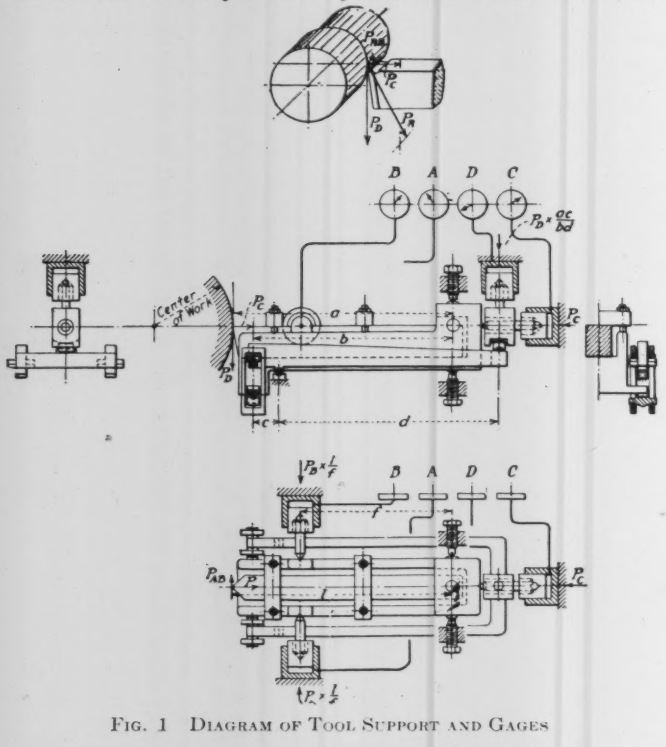

FIG. 1 DIAGRAM OF TOOL SUPPORT AND GAGES

resulted in certain improvements which have shown themselves to be of material economic value.

It was believed from the start that a study of this character could lay claim to scientific accuracy and absolute dependability only if every mechanical measurement made could, in some manner, instantly be checked. The only way in which this seemed possible was by measurement of the electric-power input into the driving motors. A check of this character, however, required a complete knowledge of all the characteristics and losses of the machine tool used. The cutting tests were therefore preceded by a most thorough study of the lathe on which the tests were conducted; and incidentally, of course, all electric meters, gages, and other measuring devices were checked as to their behavior under all possible conditions.

INVESTIGATION OF LATHE AND INSTRUMENTS

Test Equipment. The lathe used in these tests has a height of centers of about 18 in. and a distance between centers of about 118 in. It has five different spindle speeds, and in addition the speed of the 50-hp. driving motor can be varied from 350 to 1000 r.p.m. The speed of the spindle can thus be changed, in small steps, from 7 to 820 r.p.m. The feed gear has eight speeds, which will permit feeds of from 0.0037 in. to 0.1 in. per revolution. In the tests it was driven by a separate 5-hp. motor.

The tool support used in the tests was of a type which divides the total pressure on the tool into three components, a vertical component D, a surfacing component C, and a traversing component B. The tools were always accurately set at the height of the centers, in which position all measurements taken checked within $\pm 1/2$ per cent. Fig. 1 shows this support and its gages diagrammatically. A recording gage was also connected which could record all three components at the same time; it was interconnected with a recording wattmeter showing motor input in such a manner that both could be started and stopped by the same operation, so that a perfect check on readings could be had.

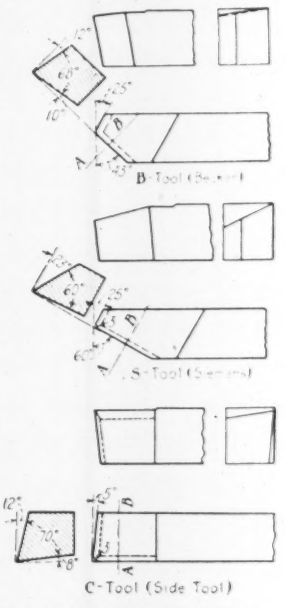

FIG. 2 SHAPES OF TOOLS USED

All tests and calibrations of tool-support instruments, gages, and motors were conducted in the most painstaking manner, and were carried to a point where all possible errors were eliminated.

Power Consumption of Feed Gear. The power consumption of the feed gear, running idle, was determined at each gear ratio and with different motor speeds. It proved extremely small as compared with the power requirements of the main spindle, as was to be expected.

In the tests the total power consumption of the feed gear was taken as motor input times motor efficiency, the latter having been accurately determined for all conditions of load and speed.

Losses in Headstock. Careful tests by means of a Prony brake with a cast-iron disk mounted on the faceplate of the lathe gave exact data with regard to the relation of actual power developed on the faceplate and the electrical input.

CUTTING TESTS

Program of Tests. The tests were made on different materials, namely, wrought iron, cast iron, and chrome-nickel steel, with three different forms of tools shown in Fig. 2, using different cutting speeds and many different chip sections.

It was aimed to determine the influence, upon the cutting forces, of

1 Cutting speed
2 Chip section
3 Shape of tool
4 Characteristics of materials (wrought iron, cast iron and chrome-nickel steel).

Tools Used. Fig. 2 shows the three tools used in these tests. The tool designated as tool B (or Becker) is one which was used by Schlesinger and Kurrein in their tests in 1913 to determine the life of tools. The angles here used represent averages as customary in machine shops abroad. The angle of 68 deg. is the one considered most favorable by Taylor. Tool S (or Siemens) is one favored by

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some manufacturers. Tool *C* (or side tool), is a form of tool having a large angle at the point, i.e., a tool with a point capable of resisting destruction under very severe conditions.

Influence of Cutting Speed on Chip Pressure. Fig. 3 shows a number of test results at various cutting speeds for wrought iron, cast iron, and chrome-nickel steel. Generally speaking, there is a tendency for chip pressures to decrease slightly with increasing cutting

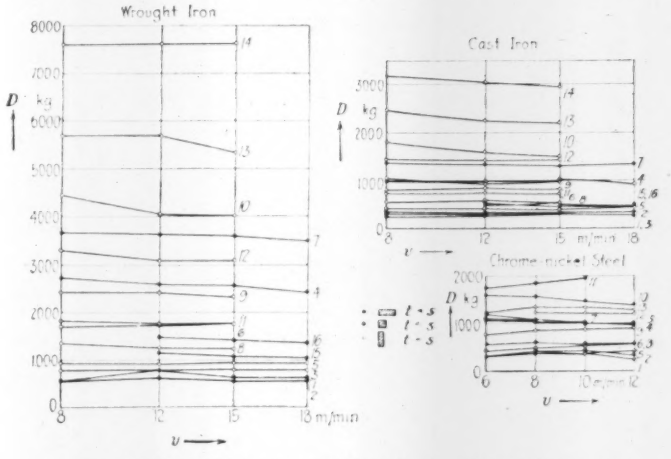


FIG. 3 CHIP PRESSURES AT VARIOUS CUTTING SPEEDS

speeds. This conclusion agrees with that reached by Taylor and by Nicolson.

Tests, not shown in these curves, at 197 ft. per min. show only a slight decrease in pressure as compared with tests under similar conditions at 59 ft. per min.

Influence of Chip Section on Chip Pressure. Before proceeding with a discussion of the influence of the chip section and of the form of tool, attention should be called to the difference between the nominal section (i.e., depth of cut × feed) and the actual section removed. The relation between the two depends on feed, depth of cut, form of tool, and angle under which the tool is applied.

Fig. 4 shows a longitudinal section through a shaft after it has been turned off, on the basis of a certain shape of tool and feed. Assuming that the shaft had previously been turned with a small enough feed so that its surface was practically smooth, it will at once be apparent that the chip removed is smaller than its nominal cross-section $t \times s$.

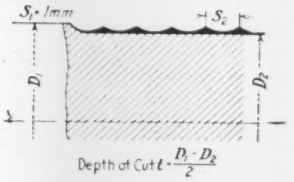


FIG. 4 SECTION OF SHAFT AFTER TURNING

The actual area removed may be expressed as a percentage of the nominal chip section, and a curve may be drawn giving this percentage for various nominal chip sections on the basis of a certain tool, a certain angle of application, and a certain feed.

Some of the results obtained in the many tests made are given in Fig. 5, which shows the power requirements for chips of certain corresponding cross-sections, all comparisons being based on nominal cross-sections. It will be seen that a chip of 1-mm. depth of cut and 10-mm. feed will require only about one-half the actual power as compared with a chip of 10-mm. cut and 1-mm. feed. Fig. 5 also shows the distribution of losses in the lathe for four particular chip sections, namely, 5 x 10 mm., 10 x 5 mm., 2 x 10 mm. and 10 x 2 mm., these dimensions also being nominal.

In the succeeding curves, shown in Fig. 6 and following, all pressures relate to actual chip sections, not to nominal ones, and the losses in motor and lathe are not shown. However, all results taken with the dynamometer support have been checked on the basis of motor input and losses in motor and lathe, as previously established. In the curves the vertical pressure components are given for the different forms of tools and for the three different materials, and the effect of increasing chip sections on this component, and therefore on the power requirements, will be clearly visible. It will be seen that with increasing chip sections the power requirements do not follow the same law with different materials. In all cases, however, the curves show that power requirements do not increase proportionally to chip section but more slowly. Attention should here be called to the fact that in Fig. 8, discussed later, the shallow chips show consistently lower power requirements than do the deep chips. This observation, already referred to, agrees with the results of other investigators, particularly Taylor, who made tests with cuts up to 1 in. deep.

Influence of Shape of Tool on Chip Pressure. In Fig. 6 are given three curves based on results with tool *B* and covering three materials. Individual test results are shown for the other two shapes of tools. Cutting speeds were about 40 ft. per min. with cast and wrought iron and about 26 ft. per min. with chrome-nickel steel. The results obtained do not reveal any material difference in the behavior of the three tools used.

In Fig. 7 are given surfacing and transverse components for the three forms of tools and the three different materials. The ordinates in each case show the relation of the respective component in relation to the vertical component (D); in other words, 0.1 means that

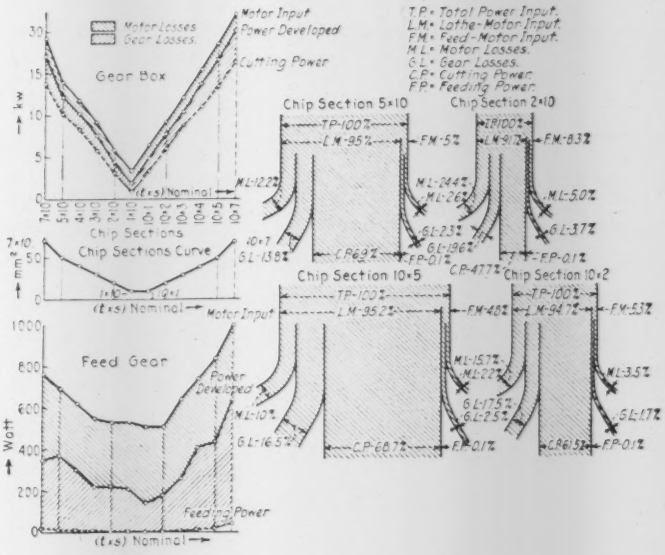


FIG. 5 POWER REQUIREMENTS AND LOSSES FOR TWO SETS OF CORRESPONDING CHIP SECTIONS

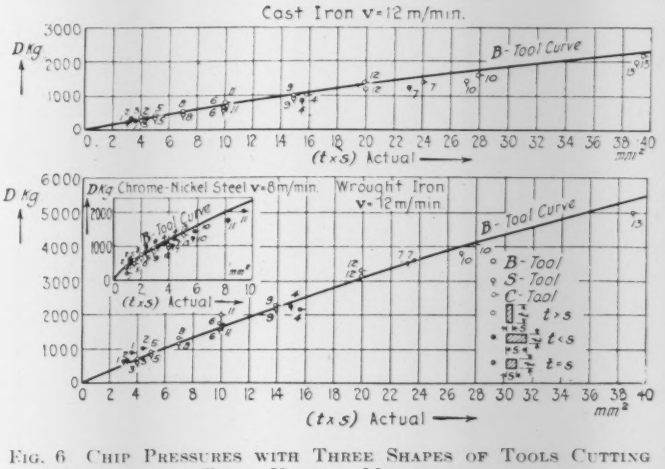


FIG. 6 CHIP PRESSURES WITH THREE SHAPES OF TOOLS CUTTING THREE KINDS OF MATERIAL

(Results with *B*-tool in solid lines; others given individually.)

the particular pressure is one-tenth of the corresponding vertical component. In all tests the *B* curve (transverse component) was highest for deep chips and lowest for shallow chips. With the latter ones, in fact, the *B* curve is sometimes located below the axis of abscissas, indicating that the remaining material on the shaft exerts a pressure in the direction of the feed. The B_1 curves refer to chip sections with t/s greater than 1, the B_2 curves to those with t/s equal to 1, and the B_3 curves to those with t/s less than 1.

Influence of Characteristics of Materials on Chip Pressure. Reference should be made to Figs. 6 and 7 which give results with the three different materials. Fig. 8 shows vertical pressures *D* and

specific vertical chip pressures (D_{sp}) for the three materials. The curves represent average results, but reference has previously been made to the location of points above and below these averages depending on the particular shape of the section.

Fig. 9 gives, on the basis of nominal chip sections, comparative results obtained by several other investigators and the author. There is also given a curve properly corrected to cover actual chip sections. Fair agreement exists among all the test results.

In an attempt to reduce chip pressures to a formula, the author followed the method adopted by Meyer in 1908 in drawing Brinell-hardness curves on double logarithmic paper, using as the abscissa the area of the Brinell impression and as the ordinate the pressure applied. He drew similar curves with the vertical component of the chip pressure as ordinate and the actual cross-section of the chip as abscissa. The curves under the circumstances became straight lines, the same as those drawn by Meyer. This at once suggested a comparison of the two sets of curves for the three materials used in the tests. When the hardness curves were added, it was found that they formed a larger angle with the axis of abscissas, and that they intersected with the chip-pressure curves. (See Fig. 10.)

There is an interesting difference in the relative location of the Brinell-hardness curves and of the three chip-pressure curves. The Brinell curve for cast iron is between those for chrome-nickel steel and wrought iron, while in the case of chip pressure, the curve for cast iron is considerably below both of the other two curves. This shows that cast iron, notwithstanding its great hardness, is comparatively easy to work on a lathe or planer.

These observations permit the determination of chip pressures and therefore power requirements in the turning and planing of materials of which the Brinell-hardness characteristics are known.

The Process of Metal Cutting

The Formation of a Chip. It is surprising that the actual formation of a chip has been studied but little, with the result that rather vague ideas still prevail regarding this fundamental operation. A clear observation of a turning tool working on a lathe is practically impossible, since, during the process of turning, the point is imbedded in the material to such an extent that the exact action of the tool cannot be seen. The author therefore decided to use a shaping machine and a work piece which was planed and finished on one side so that all fissures and all compression of material could be well observed. The tool was set so that it worked on this finished surface. Since, however, in a shaping machine the tool moves, and since it appeared desirable to photograph individual phases of the process, a planer was later substituted which was operated by hand so that it could easily be stopped at any point desired.

The two sets of pictures shown in Fig. 11 refer to two different sizes of chips, but they both show about the same successive steps. Referring first to the left row, it will be noticed that in the top picture the point of the tool compresses a small quantity of material. In the second picture this compression has gone so far that a large tear has been caused between the main part of the work piece and the chip element being separated from it. This tear extends from the point of the tool to a point on the surface of the work piece a small distance in front of the tool. In the succeeding pictures the process described repeats itself. In the fourth and fifth pictures of the left-hand row small fissures are noticeable under the point of the tool, a phenomenon to which, as far as the author knows, no attention has been called heretofore.

The second row of pictures shows similar steps, but more clearly since a larger chip was being taken and since the photographed steps follow each other more closely.

In order to show that chips in turning and planing have the same characteristics, both kinds were photographed in plan view and side view (see Fig. 12). The similarity of both chips leaves no doubt that the process in turning is the same as that in planing, notwithstanding the fact that the two chips were taken at entirely different cutting speeds. The surfaces from which the chips were cut also show a similar appearance. The lower picture in Fig. 13 shows the surface from which one of the sets of chips was turned at a cutting speed of about 27 ft. per min., while the upper one shows the surface obtained in the test on a planer. Both pictures show very similar lines across the surface. The cutting edge of the planer tool used was inclined at an angle of 5 deg. to a line vertical to the direction of motion, and the fissures in the corresponding surface show a similar direction. The individual fissures indicate the points at which the cutting edge started a new operation after a chip element has been broken off, or probably a point slightly later than this starting point.

From what has been said, it would seem that the process of cutting might be divided into four parts:

1 The initial compression of material in front of the tool. This action causes the tool to get its first new grip on the work piece after a chip has been broken off.
2 A slight initial tear is caused between the main part of the work piece and the material which is to form the next chip element.
3 The material which is to form the next chip element is compressed and bulges on the lip surface.
4 A tear is caused between this material and the main work piece. This tear is generally not complete but, with the exception of cast iron and similar materials, a certain connection remains between the individual chip elements. This connection makes a chip of the individual elements broken off.

The conception here outlined explains the oscillating form of chip-pressure curves obtained by Nicolson and others. In these curves the points of maximum pressure represent the moment of maximum compression of the chip elements, while the lowest points immediately succeeding the points of maximum pressure represent the moments after the chip elements have been broken off.

Structural Changes in the Chip. Professor Kurrein's microscopic observations in 1905 have been confirmed and materially amplified by the author. From the very large number of microscopic pictures taken to investigate the structure of a chip, only one is given here. Fig. 14 shows a vertical section parallel to the motion of the tool and taken at the root of a chip, i.e., in the zone directly in front of the point of the tool. The effect of the tool, located at the left-hand side of the picture and moving to the right, upon the material not yet severed is clearly visible. The difference in appearance of the part which will next be detached and the part which remains on the work piece is very marked, and the line of separation at once apparent.

Effect of Chip on the Tool. Aside from the effect of the turning process upon the cutting edge proper, consisting mainly in destruction by impurities contained in the material, there is a gradual effect produced by the chip on the tool which in time affects the cutting edge. This latter action is radically different from the former; it does not start at the edge but on the lip surface a slight distance back of the edge, and causes a cavity which in time becomes larger, extending more and more toward the rear but also slowly approaching the cutting edge. When it comes in contact with the latter it breaks it down completely at the point reached. This process will become entirely clear by reference to Fig. 15 in which 1, 2, and 3 show the successive stages of destruction.

The phenomenon mentioned is most noticeable with heavy chips, but even with small chip sections it can be observed, especially when turning material of great toughness. Fig. 16 shows the lip surface of a tool used for chrome-nickel steel with chips of only 1 mm. depth and 3 mm. feed. The result is very marked. The cavity begins about a sixteenth of an inch back of the cutting edge, but the edge itself is not as yet injured. This important action of the chip has been given surprisingly small attention heretofore. It is clearly a result of the great compression of the chip element in formation and of the friction between the chip and the lip surface upon which it slides. With larger chips, as mentioned before, the effect becomes even more marked, and incidentally the distance from the cutting edge at which the cavity starts is slightly increased. With wrought iron this effect is less marked; with cast iron it is not noticeable.

A New Type of Cutting Tool

As a result of these observations a new form of tool has been designed which is intended to reduce to a minimum the amount of compression and deformation of the chip and the marked cup action on the lip surface coincident with it. This new form of

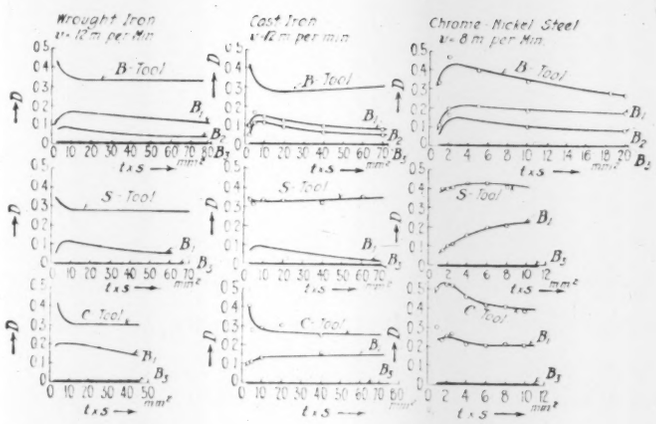


FIG. 7 TRANSVERSE COMPONENTS B_1, B_2, B_3, AND SURFACING COMPONENTS (C) FOR THREE TOOL SHAPES (B, S, C) AND THREE KINDS OF MATERIAL CUT

(The ordinates represent decimal parts of the corresponding vertical component D. B_1, B_2, B_3, respectively, correspond to chip sections with t/s greater, equal to, or less than unity.)

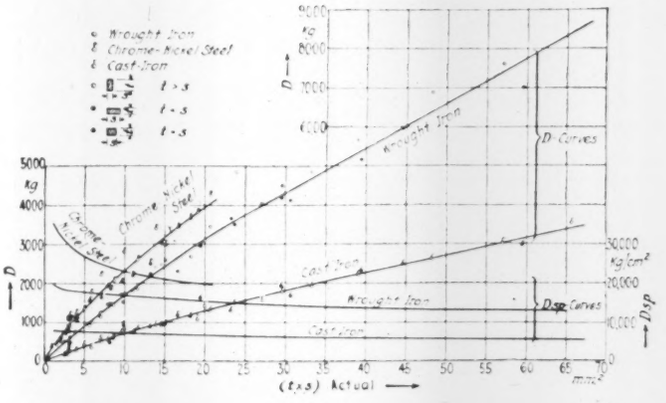


FIG. 8 VERTICAL PRESSURES (D) AND SPECIFIC VERTICAL PRESSURES (D_{sp}) WITH THREE MATERIALS

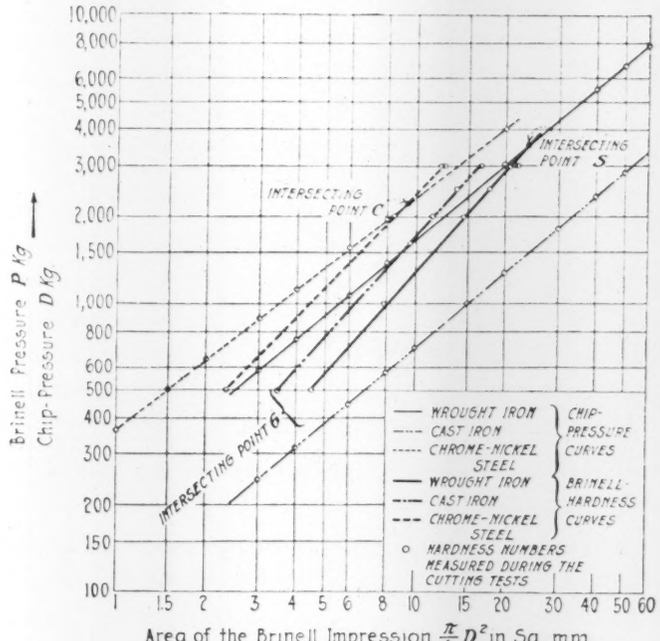


FIG. 10 BRINELL-HARDNESS CURVES AND CHIP-PRESSURE CURVES FOR THREE MATERIALS

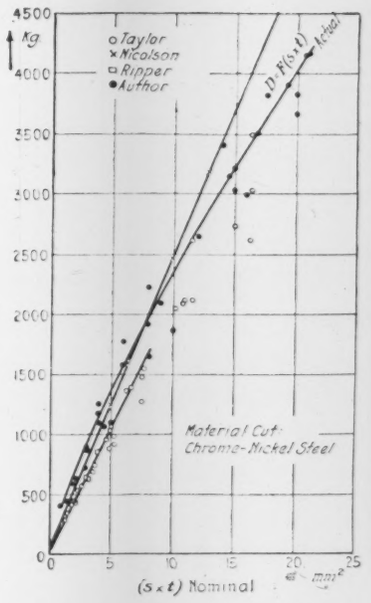


FIG. 9 COMPARISON OF CHIP PRESSURES AS MEASURED BY VARIOUS INVESTIGATORS

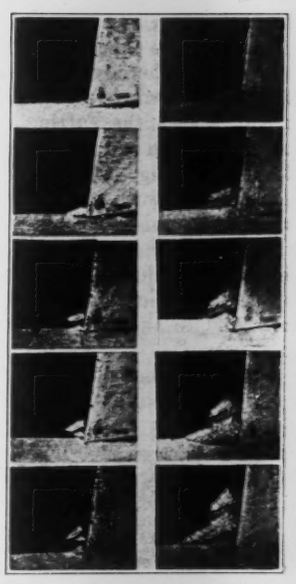
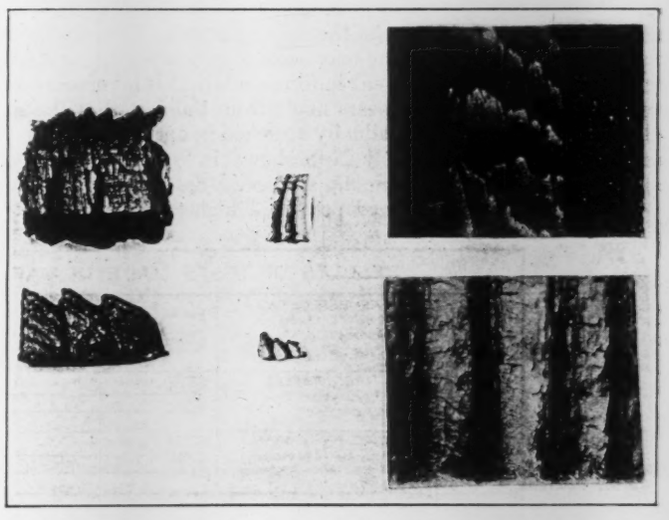


FIG. 11 SUCCESSIVE STEPS IN THE FORMATION OF CHIP ELEMENTS



FIGS. 12 AND 13 COMPARISON OF LATHE AND PLANER CHIPS

(The surfaces from which the chips were taken are shown at the right.)

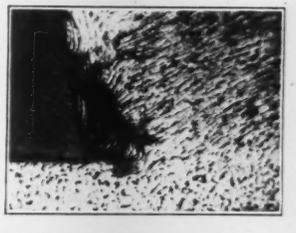


FIG. 14 SECTION AT ROOT OF CHIP

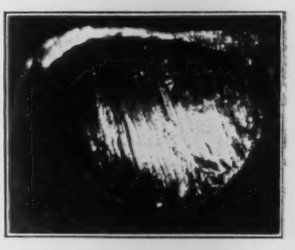


FIG. 16 LIP SURFACE GUTTERED OUT BY THE CHIP

FIG. 15 EFFECT OF CHIP ON LIP SURFACE

FIG. 17 KLOPSTOCK CUTTING EDGE

tool embodies the depression in the lip surface produced in time by the chip. The expensive and detrimental work of the chip on the lip surface is thereby kept down to a minimum and the compression of the chip reduced materially. Fig. 17 shows one form of tool provided with the cutting edge developed by the author. The tool here shown has a front and rear cutting edge and has along both of these cutting edges a narrow raised strip projecting above the rest of the lip surface. The connection between the depressed section and the raised strip is made by means of a fillet, with the result that the cutting edge has a strong support and that the heat is efficiently conducted away from the cutting edges. The lip angle is generally chosen in the neighborhood of 70 deg. which also is favorable to the conduction of the heat away from the cutting edge proper.

From what has been said before, the reduction in the chip pressure upon the lip surface, which has been definitely proven, will not appear surprising. It is furthermore believed that the favorable action of the raised strip upon the chip element in formation takes place due to the fact that the rear edges of the raised strips exert a pressure upon the chip element in formation that is favorable to its separation. This effect appears likely from the fact that the rear edge of the raised strip is gradually attacked in operation and wears away, just as has previously been shown in connection with the cutting action in a continuous lip surface. (See also Fig. 18.)

Fig. 18 Wear of the Klopstock and of the Standard Tool

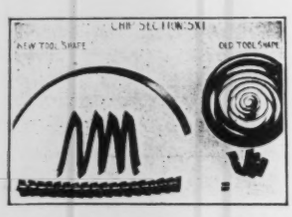


Fig. 19 Chips Cut with Klopstock and with Standard Tools

The characteristics outlined permit of higher cutting speeds or larger chips ections, or both, than are practicable with the standard tools now used. As a result of this, production is increased; and this is done without straining the machine tool used, since even with heavier chips the chip pressure is not increased and since with the same size of chip the chip pressure is reduced, so that even a higher cutting speed will not mean greater power consumption.

Extensive tests have been made with many different and even abnormal chip sections in order to test the working and the life of this cutting edge. The results obtained vary with conditions as was to be expected, but as an average it may be stated that with chrome-nickel steel the vertical component is reduced about 26 per cent and with wrought iron, 14 per cent. The transverse force is reduced on an average about 65 per cent, and the surfacing force 60 per cent with chrome-nickel steel and 50 per cent with wrought iron. The difference between the vertical component and the total chip pressure is only about 2 per cent.

A noticeable difference exists with regard to the exact saving in power consumption with different chip sections. With chips causing a considerable amount of material to be taken off by the rear cutting edge, the material flowing over this rear cutting edge meets the material coming from the front cutting edge and a certain amount of interference and deformation takes place as a result thereof. These are the greater the smaller the angle between the front and rear cutting edges, and an angle less than 90 deg. is therefore inadvisable. On the other hand, a chip section which causes material to be cut almost entirely by the front cutting edge will allow the chip to flow freely without interference. The chip sections of great depth and small feed therefore show a greater saving than those of small depth and large feed.

The difference in the manner of wear in operation is clearly shown in Fig. 18, which represents two tools made of the same tool steel and operated under similar conditions—in this case, abnormal conditions. The cutting speed was about 197 ft. per min. The standard tool illustrated at the right shows the edge near the point destroyed, while the tool with the raised strip shows destruction of a short section of the narrow strip which is clearly seen to have started in the rear. The point in this latter case is still in fair condition.

Fig. 19 shows the different appearance of chips produced by standard tools and the new tool. The right-hand column of Fig. 19 shows chips produced with a standard tool. The spiral at the top represents a chip cut with a newly ground tool; the picture below shows a chip with the tool beginning to get dull, and the small piece at the bottom represents the sections torn off with the dull tool. The time during which this last kind of chip was produced in these particular tests represented more than half that during which the tool was used, and practically during the whole period when short chips were produced a chattering noise was heard.

The left-hand column shows chips produced with the new tool. At the beginning chips are produced of a very large radius. This radius gradually decreases as the cutting edge is affected, and still later a long helix of small radius appears. In most cases this shape of chip continues until finally the fact that the tool has become dull is indicated by the chattering noise it makes. With the same depth of cut and with the same feed the chip produced by the new tool has a very much smaller cross-section than that produced with the standard tool, which is of course due to reduced compression and deformation in the case of the chip produced by the new tool. It goes without saying that in the latter case the chip runs off at a correspondingly higher speed.

An extremely interesting fact in connection with this is that the chips produced by the standard tools are of dark color, while those produced by the new tools under exactly the same conditions

RESULTS OF TESTS MADE IN VARIOUS GERMAN RAILROAD SHOPS

HALLE R.R. SHOPS	DULLNESS-TESTS OF TOOLS a) WITH WHEEL AXLES The Klopstock-Tool has shown a 5-times greater Life and a 10 Per Cent Increase in Production Simultaneously	Standard-Tool Life / Klopstock-Tool Life 440 2200 Length Turned in mm	DRESDEN R.R. SHOPS	DULLNESS TEST OF TOOLS WITH CAR WHEELS Life of Klopstock-Tool 3-4-times greater	Standard-Tool Life / Klopstock-Tool Life 2 6 Pieces Turned
	b) WITH TENDER WHEELS Life of Klopstock-Tool 6-times greater	Standard-Tool Life / Klopstock-Tool Life 1.5 6 Pieces Turned	OPPELN R.R. SHOPS	I- DULLNESS TESTS OF TOOLS (WITH WHEELS) Life of Klopstock-Tool about 3-times greater	Standard-Tool Life / Klopstock-Tool Life 5.5 10 Pieces Turned
	c) WITH LOCOMOTIVE WHEELS Life of Klopstock-Tool 6-times greater or a 50 Per Cent increase in Production	Standard-Tool Life / Klopstock-Tool Life 0.3 2 Pieces Turned		II- POWER-CONSUMPTION TESTS 1) TEST WITH ONE TOOL Economy in Power Consumption with Klopstock-Tool 33⅓ Per Cent	Standard-Tool Power Consumption / Klopstock-Tool 4amp 6amp Current Used Economy in Power
HARBURG R.R. SHOPS	DULLNESS TESTS OF TOOLS a) WITH CAR AXLES Life of Klopstock-Tool more than 7-times greater	Standard-Tool Life / Klopstock-Tool Life 400 3385 Length Turned in mm		2) TEST WITH TWO TOOLS Economy in Power by using Klopstock-Tools about 27 Per Cent AVERAGE OF BOTH TESTS 30 Per Cent Economy	Standard-Tool Power Consumption / Klopstock-Tool Current Used 11amp 15amp Only Current for Cutting is Given Economy in Power
	b) TURNING OF WHEEL TIRES Life of Klopstock-Tool 4.4-times greater	Standard-Tool Life / Klopstock-Tool Life 235 1045 Length Turned	GRUNEWALD R.R. SHOPS	COMBINED DULLNESS AND POWER-CONSUMPTION TESTS TURNING OF A CAST-STEEL AXLE The Life of the Klopstock-Tool has been 3.25-times Longer and the Power Consumption was 25 Per Cent Lower	Standard-Tool Life / Klopstock-Tool Life 100 325 Length Turned in mm
	c) BORING OPERATION ON TIRES Life of Klopstock-Tool 2-times greater	Standard-Tool Life / Klopstock-Tool Life 285 330 Length Turned			Standard-Tool Power Consumption / Klopstock-Tool 5amp 20amp Current used Economy in Power
	d) WITH LOCOMOTIVE WHEELS Life of Klopstock-Tool 4-times greater	Standard-Tool Life / Klopstock-Tool Life 0.5 2 Pieces Turned			

Fig. 20 Relative Life of Klopstock and Standard Cutting Edges

are very light in color. The difference in temperature has under certain conditions been found to be as much as 100 deg. cent. or 180 deg. fahr.

The material reduction in the heat produced, and the efficient manner in which this heat is conducted away from the cutting edge, leave the latter in a good condition for work up to the last, with the result that the surface turned by means of the new tool is generally better than that turned with the standard tool.

A most important point with any tool is, of course, its life. A large number of tests were made to determine the relative life of the new and standard types of tools. It was invariably found that under the same conditions the new tool lasted very much longer than the standard tool, which no doubt is a result of the combination of a large lip angle with a reasonably low temperature in operation. In other words, the new tool has a cutting edge which is mechanically strong on account of its large lip angle, but which remains reasonably cool, first, because the heat produced is less, and, second, because the design of the tool does not interfere with efficient conduction of the heat away from the cutting edge.

Fig. 20 presents the results of some of the tests made in various German railroad shops. Hundreds of similar tests have been made which have given similar results. The conditions prevailing in the tests here tabulated were exactly the same in the case of both tools, and only the relative life of the cutting edge was compared.

The manner of producing the new form of tool has been given careful study. Complete tools or cutting tips for welding to shanks are generally made in dies by hammering or pressing. If made in this manner, the tool need only be ground along the narrow raised strip and on the clearance surface. The depressed part of the lip surface is in this case left untouched. No difficulties are encountered in the tool room; in fact, it is possible with the new type of tool to reduce materially the number of forms of tools. Foreign shops have found it entirely practicable to get along with a very few standard forms, owing to the fact that cutting conditions are more favorable with the new tool.

In normal operation the cutting edge can be reground by the operator four times, after which it must be newly forged. The tool is of course hardened at the same heat at which it is forged, i.e., it is first forged and then its temperature is brought exactly to the proper hardening temperature.

If it is desired to produce the depression in the lip surface entirely by grinding instead of forging, this can readily be done by means of a special attachment to a grinder equipped with a small-diameter grinding wheel. The attachment permits of easy adjustment of the tool to the proper angles, and the regrinding of a depression which has become too shallow can, by means of this attachment, be accomplished within a few minutes.

In estimating the effect of reduced chip pressures upon the amount of work which can be turned out on a particular lathe, the following should be kept in mind:

1 A decrease in chip pressure of, say, 20 per cent will permit an increase in the chip pressure of 25 per cent before the same chip pressure is again reached

2 A permissible increase in the chip pressure of 25 per cent means an increase in the area of the chip amounting to considerably more, the exact amount depending on the character of the material turned. On the other hand, an increase in the chip area of 45 or 50 per cent does not mean an increase in the production by a corresponding amount, since the time for setting up and removing the work piece must be taken into account.

Generally speaking, results obtained in foreign shops have justified the belief that production can be increased about 30 per cent by means of the new tool.

Principles of Metallurgy of Ferrous Metals for Mechanical Engineers

II—The Crystalline Structure of Ferrous Metals[1]

BY LEON CAMMEN,[2] NEW YORK, N. Y.

SUBSTANTIALLY all ferrous metals are crystalline in structure. The impression prevalent at one time that wrought iron is fibrous is due to superficial observation which shows streaks of metal interspersed with streaks of slag. Another impression, namely, that repeated shocks, vibration, or the application of excessive alternating stresses produce "crystallization" in metals is likewise erroneous as all ferrous metals are crystalline to start with. It will be shown later, however, how repeated alternating stresses do cause weakness by affecting the crystalline structure of the metal, though not by creating it. Because of the fact that many mechanical and chemical properties of ferrous metals depend on the state of crystallization, this subject assumes considerable importance to the mechanical engineer.

CRYSTALS IN GENERAL

The atomic theory of today teaches us that in a solid body the atoms or molecules tend to move in closed paths about fixed equilibrium positions. These equilibrium positions may possess a special regularity in arrangement or may not. If they do, the body is called "crystalline;" otherwise it is classed as "amorphous." A crystalline body throughout which this order is undisturbed constitutes a single crystal. In a perfect crystal the same symmetry characteristics are maintained, no matter what may be the absolute size of the specimen. Thus, if a large rhombohedral fragment of calcite is split into bits by successive blows it will cleave into many minute rhombohedrons, each of which possesses the same symmetry properties as the original piece.

The fundamental symmetry characteristics of crystals are shown by a study of the distribution of their plane faces and of various physical properties, such as refractive indices, which are unlike in different directions, and also by the behavior of extremely short (X-ray) waves when passing through crystalline bodies. In this way certain elements of symmetry have been recognized as present in crystals. The first and most important are the axes of symmetry. If in a figure a rotation along some line combined with translation along its length leaves the figure in its original geometrical aspect, the figure is said to have a "screw" axis of symmetry. But if the amount of the translation is zero, the symmetry axis is a simple rotational axis of symmetry. In crystals only certain axes are observed.

Next come the planes of symmetry, a plane of symmetry being defined as any plane that can be passed through a figure so that every point upon one side of the plane has a corresponding point on the other side in a mirror-image relation to it.

There are also centers of symmetry in crystals, a point being a center of symmetry if a line drawn from any other point of the figure to it and extended an equal distance beyond encounters a point corresponding to the arbitrarily chosen point.

Investigation has shown that there are but thirty-two distinctive ways of taking these elements of symmetry singly and of combining them together. From this a number of groups have been built up, each of which represents a system or class of crystals. In fact, the whole important science of crystallography has been reared on this foundation. However, only certain comparatively simple crystal structures are encountered in the case of commercially known metals.

[1] Second of a series of articles discussing the underlying physical and chemical processes involved in the metallurgy of iron and steel. The first, dealing with the physico-chemical properties of iron alloys, appeared in the May issue, p. 339.

[2] Associate Editor, MECHANICAL ENGINEERING.

THE INTERNAL STRUCTURE OF CRYSTALS

Such elements of crystal structure as the axes, planes, and centers of symmetry deal with the symmetry exhibited by a crystal as a whole. When we come to the subject of the internal symmetry of crystals, which is the question of the special grouping of symmetry elements, it is found that these are affected by the symmetrical patterns in which the groups of points are distributed. These symmetrical patterns are called "space lattices." A space lattice consists of a series of points in space, such that every point is situated symmetrically to every other point; space being usually imagined as divided into cells by three sets of parallel planes, with the planes of each set at equal distances from each other and parallel to each other.

The distances between planes may be different in the various systems and the three sets of planes may form any angles with each other. The points of intersection of these planes constitute a space lattice. Taking, for example, Fig. 6, we have here a portion from a "body-centered"[1] cubic space lattice and the cube *OABCD-EFG* is a unit cell. For many purposes, including all those of crystal analysis, the unit cell may be taken as the smallest volume which will build up the entire grouping when repeated indefinitely by simple translations along the axes of coördinates.

The space groups define the positions that can be occupied by the atoms in crystals. If an atom is located at a certain point which we may call *XYZ* within a crystal, there must be exactly similar atoms at each of the equivalent points within the unit domain of its corresponding space and at the equivalent points arising from a repetition of this unit along the axes of reference. Measurement of the X-ray diffraction effects from crystals seems to show that the number of chemical molecules associated with the unit cell is usually small, and means have now become available to determine the number and location of the atoms within the crystal structure.

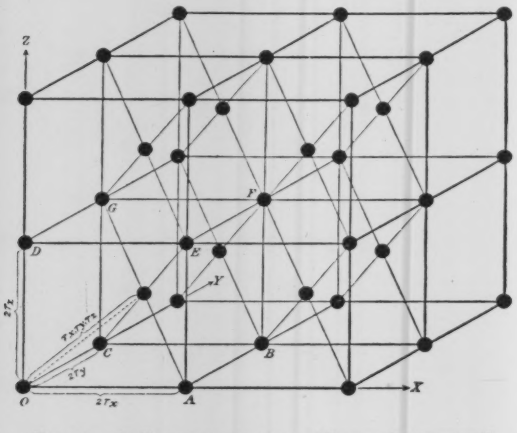


FIG. 6 A PORTION FROM A "BODY CENTERED" CUBIC SPACE LATTICE

(The cube *OABCDEFG*, with which are associated two points in the lattice, is the unit cell. Reproduced from Fig. 48, p. 59, of the Structure of Crystals, by Ralph W. G. Wyckoff, American Chemical Society Monograph Series, by courtesy of the publishers, Chemical Catalog Company, Inc., New York City.)

The arrangement of atoms in metal crystals appears to be simple. The simplest is that of a cube with an atom at each corner. This is what is known as the "simple cubic arrangement." No metals have been found to possess this arrangement, but it is encountered in salts composed of equal numbers of positive and negative ions such as the alkali halides. The arrangement encountered in metals, iron in particular, is that of a cube with atoms at the corners of a system of equal, closely packed cubes and an additional atom either in the center of each cube face or in the center of the cube itself. The former is known as the "face-centered cubic arrangement" of atoms and is illustrated at the right in Fig. 7, which represents the crystalline structure of gamma iron according to Westgren and Lindh;[1] the latter is the "body-centered cubic arrangement" of atoms and is shown at the left in Fig. 7, representing the crystalline structure of alpha and beta iron according to the same authorities. The face-centered cubic arrangement may be considered as formed by two simple cubic lattices intersecting each other in such a manner that the corner atoms of one lattice fall at the centers of the faces of the other lattices, so that any particular set of atoms may be equally well regarded as corner atoms or as face atoms. Each atom is surrounded by twelve equidistant and symmetrically situated atoms. This arrangement, illustrated in Fig. 8, is one of two which furnishes the closest possible packing for a number of equal cubes. It is sometimes referred to as the "cubic close-packed arrangement." In the body-centered arrangement illustrated in Fig. 9 (Hull) the corner atoms and center atoms are interchangeable, so that if the system of lines of Fig. 9 had started with one of the center atoms, all the corner atoms would become center atoms, and vice versa. Each atom is surrounded by eight equidistant and symmetrically situated atoms. This arrangement is not so closely packed as the face-centered arrangement.

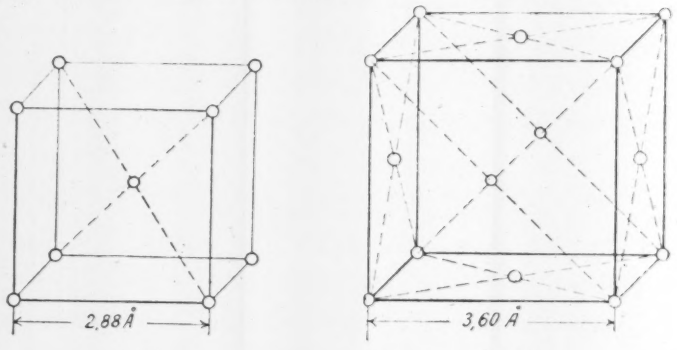


FIG. 7 CRYSTAL STRUCTURE OF ALPHA AND GAMMA IRON (WESTGREN AND LINDH)
(*Eisen* = Iron.)

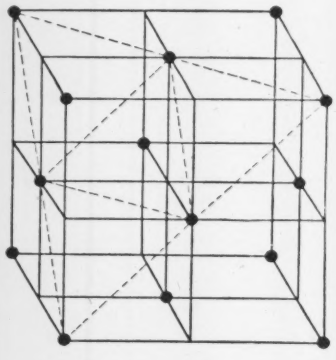


FIG. 8 FACE-CENTERED CUBIC ARRANGEMENT OF ATOMS (HULL)

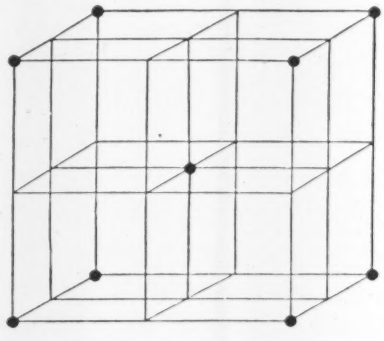


FIG. 9 BODY-CENTERED CUBIC ARRANGEMENT OF ATOMS (HULL)

From various investigations it would appear that iron at room temperatures has what is known as a body-centered cubic arrangement of crystals, essentially such as shown in Fig. 9 and at the left in Fig. 7. From what has been said in Chapter I on the physico-chemical properties of iron, it would appear that this corresponds to the structure of alpha iron. Measurements (Westgren[1]) have also shown that the cubes measure 2.91 Ångstrom units (each such unit is 10^{-8} cm. on a side). A similar investigation has shown that iron in the beta range of temperature has the same space lattice, i.e., body-centered cube, as alpha iron, but that the side of the elementary cube is slightly larger, which can be accounted for by the expansion of iron at the temperature of 830 deg. cent. at which the beta lattice was measured. It may be pointed out in this connection that the difference between the properties of alpha and beta irons is so slight that for a long time there has been a good deal of doubt as to whether two such allotropic modifications really existed. There has never been much doubt as to the existence of the gamma-iron allotropic modification, as the properties of gamma iron are clearly different from those of alpha or beta iron. This is very well accounted for by the observation of the

[1] This term is explained later.

crystal of gamma iron, as at 1000 deg. cent. it has a face-centered cube somewhat larger than that of alpha iron, namely, 3.61 Ångstrom units.

A fact, not new but none the less remarkable, has been apparently established by an investigation made by Dr. Arne Westgren and G. Phragmen, who found that when austenite is heated through the uppermost critical point (about 1400 deg. cent.) it undergoes another modification (into what is designated as "delta iron"), and that this modification, like alpha iron, is body-centered.

In Chapter I reference was made to allotropic modifications of iron, or iron that without changing its chemical constitution appears in materially different forms as alpha iron at one time, gamma iron at other times, and possibly beta and delta iron under still different conditions. A glance at Fig. 7 will easily explain why this happens, alpha iron being body-centered and gamma iron face-centered cubic crystals without apparently any change taking place in the atoms constituting the different arrangements. According to Westgren and Lindh,[1] high-speed tool steels hardened at 1275 deg. cent. have practically the same crystalline structure as alpha iron. As regards the structure of martensite, of importance in hardened steels, a recent investigation by K. Heindlhofer[10] would appear to show that martensite has a body-centered lattice like that of alpha iron, but the lattice is contracted about 0.4 per cent on a side and is also distorted. This distortion disappears largely, however, after heating the martensite at 260 deg. cent. for 30 minutes, although the hardness is not much decreased. From a comparison of the densities of specimens of martensite with those of alpha iron and of cementite, it would appear that martensite is a mixture of (1) a solid solution of carbon in iron, the C atoms replacing the Fe atoms, and (2) of finely dispersed cementite, the former being unstable even at room temperature. Both seem to contribute to the hardness, the solid solution by distorting the lattice, and the cementite by being so fine-grained.

Practical Applications of the Knowledge of Crystalline Structure of Metals

At first glance it would appear as if the subject of crystalline structure of metals had merely a theoretical interest. After all, it simply shows the distribution of atoms and molecules in a metal, and one might question the utility of such knowledge to the practical engineer, presumably more interested in what the metals with which he has to deal are than in the underlying reasons why these metals are what they are. Actually, however, it is not so, for knowledge regarding the crystalline structure of metals throws a vivid light on a number of phenomena with which the engineer has to deal. It explains, for example, the reasons for the toughness and brittleness of certain metals and alloys, the phenomena of hardening, the whole subject of strength of metals, and the true meaning of the conception of solid solutions without which the metallurgy of steel cannot be properly understood. We shall start with this latter.

Solid Solutions

In the first chapter, on the Physico-Chemical Properties of Metals, it was shown how steel from the liquid state passed to that of austenite, which is a "solid solution" of carbide of iron in gamma iron, and an example was given to illustrate the meaning of the term "solid solution." An engineer, however, while knowing in a general manner that solid solutions exist, cannot well realize as a rule just how such a thing is possible. The mechanism of solutions in liquids is fairly obvious. If we drop a lump of sugar into a cup of hot tea, we can at times see how the water tears away particles of sugar, breaking them up into minute particles, and it is easy to understand how gradually, especially when the liquid is agitated, the sugar spreads through the whole cup. When we deal, however, with solid solutions, the conditions seem to be such as to oppose the operation of the process of solution. Here we are dealing with solids such as carbide of iron and gamma iron, and it does not appear at all clear how interpenetration and relative movement of particles of the two solid substances can take place to an extent necessary to produce the intimate phenomenon known as solution. In other words, it is not clear to those who have not studied the crystalline structure of metals how the diffusion between metals can occur. This is, however, quite well explained when we remember that each crystal cell of the metal consists of several atoms located symmetrically with respect to some central point at not inconsiderable distances from each other, with an additional atom in the center of the crystal or on its face.

Starting with this the British metallurgist, Dr. Walter Rosenhain, put forward a theory to explain diffusion between metals. What he claims is, that where solid solution occurs, an atom or atoms of the solvent displace corresponding atoms of the solute, the "stranger" atom being forced to take the place formerly occupied by a "host" atom and to behave in exactly the manner of its host. As a rule, the stranger atoms have dimensions different from those of the host atoms which they displace, and the substitution of a larger or smaller atom in the crystal structure sets up intense local forces in the immediate neighborhood of the stranger atoms which cause slight slips, probably of only a few atom diameters. Only a row or single line of atoms may move, but this would be sufficient to account completely for diffusion.

It has been noted in Chapter I that when steel passes through certain critical points a change occurs in the volume simultaneously or nearly simultaneously with the "kink" in the temperature-time curve. It was generally stated that both the evolution and absorption of heat, which keep the temperature of the metal steady for a while after it has been either gaining it or losing it, and the change in volume were due to changes in the structure of the metal. We can see now from our knowledge of the crystalline structure just what these changes are. If iron is heated through the upper critical range the former sorbitic, troostitic, and martensitic structures pass into austenite, which is a solid solution. This means that the molecules of carbide of iron shoulder their way into the neighboring gamma-iron lattices; the situation being practically as if a grown-up person came to a table occupied by children and after pushing one of them out of the way, sat down to the meal. The invader molecule being of different size, produces a change in the volume of the resultant material (solid solution). Not only that, but it takes a certain amount of energy for the stranger molecule to force its way into the original space lattice, as the host atom, so unceremoniously shouldered out of the way by the stranger, does not yield his place without certain opposition. The energy thus absorbed is shown negatively on the pyrometer by the fact that the rise of temperature is suspended for a while notwithstanding the fact that heat is being added the whole time. When the metal is being cooled through the critical range the contrary phenomenon takes place and the fall of temperature stops for a while, notwithstanding the fact that heat is being lost from the metal. This would indicate an internal evolution of heat produced by expelling the stranger molecules and reconstructing the original lattice. On the other hand, however, A. Westgren and G. Phragmen[11] claim that austenite is not formed by simple substitution of metal atoms by carbon atoms, but represents an additive product with the carbon atoms situated in the interstices between the metal atoms.

The Structure of Actual Metals

Such figures as Figs. 7, 8 and 9 show the structure of what might be called an "ideal crystal," such as would be formed provided there were no interferences with its formation on the part of other crystals. This happens when a single crystal is allowed to develop from a molten bath. Actually, however, in the vast majority of cases a somewhat different process takes place. As the metal cools, nuclei form at various spots throughout the molten mass. These nuclei or minute crystals grow in size with the loss of heat. As a rule this growth does not take place simply by the accretion of concentric layers, but by the crystal's acting in the manner of an octopus and throwing out a number of arms into the surrounding liquid, these arms in their turn sprouting feelers, and the final structure somewhat resembling a bush. (This is one of the reasons why such formations are called "dendrites," from the Greek word *dendron*, meaning a tree or bush.) Since, however, these formations are initiated simultaneously at a number of points throughout the liquid mass, sooner or later the branches of the various dendrites encounter each other and a fight for room begins, the mutual pressure of the branches preventing them from retaining the ideal crystal form. The result is a system of crystals with an orderly arrangement of the atoms in the interior but an

external shape determined by the influence of other crystals. Such crystals are called "crystallites" or "grains." No two adjacent grains in the cast metal have the same orientation, that is, the same direction of the crystallographic axes. If they had the same orientation, the two grains would become one. A metal, therefore, which has solidified from the molten state consists of an aggregate of differently oriented grains, somewhat like the structure diagrammatically shown in Fig. 10, taken, through the courtesy of D. Van Nostrand Co., Publishers, from An Introduction to the Study of Physical Metallurgy, by Walter Rosenhain, 2nd edition, 1923 (Fig. 14e, Plate IV).

While it is generally assumed that crystallization is a continuous process, Col. N. T. Belaiew,[4] from observation on the Tschernoff crystal (a steel crystal about 15 in. long which was found in the upper portion of the pipe in the sink head of a soft-steel open-hearth ingot weighing about 100 tons), comes to the conclusion that crystallization may proceed not continuously but by pulsations or leaps, and advances the proposition that perhaps that is a general occurrence.

The question now arises, What is the structure of the black spaces between the grains in Fig. 10? The origin of these spaces is obvious. The grain formation proceeds until all the liquid metal that can go into grains is exhausted. When this happens, and possibly when some other factors come into play, grain formation stops. The original idea was that there were voids between the grains, but this hypothesis has been abandoned, and the theory which is more and more receiving acceptance today is that the spaces between the grains are filled by an amorphous material acting as an intercrystalline cement.

Before we proceed to the discussion of this latter, it may be mentioned that the above discussion of the method of growth of dendrites explains the difference in structure between chilled castings and slowly cooled castings. (An interesting example in the difference of structure of the two may be found in photomicrographs shown by Paul Oberhoffer.[7]) It is generally known that chilled metals are fine-grained, while slowly cooled metals are coarse-grained. This is obviously due to the fact that in chilled metals the temperature drops so rapidly that an enormous number of nuclei are formed, and the dendrites are developed at so many points that there is but little room for them to grow. In slowly cooled metals the number of dendrites is comparatively small, and enough time and material are available to permit each nucleus to appropriate a comparatively large amount of liquid metal for its growth, which results in a large and coarse-grained structure.

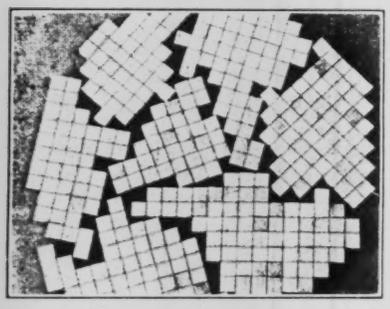


Fig. 10 Diagrammatic Illustration of the Distribution of Crystals and Intercrystalline Material

(Fig 14 of an Introduction to the Study of Physical Metallurgy, by Dr. Walter Rosenhain. Reproduced by courtesy of the publishers, D. Van Nostrand Company, New York City.)

The Amorphous-Metal Hypothesis

The term "amorphous" is in the main sense the direct opposite of "crystalline" and defines materials whose molecules do not possess such a regularity of arrangement as characterizes crystalline materials. In application to metals Beilby[3] used the qualifying adjective "vitreous," applying it specifically to substances which in some degree resemble the glasslike form assumed by the silicates when they are solidified from the molten state. Amorphous solids are essentially undercooled liquids of great viscosity. The constitution of the liquid is preserved, and with it the property of fluidity masked from casual observation by the high viscosity. (Jeffries and Archer, p. 64.) Glass may be taken as an example. On cooling from the molten state it passes in a continuous manner from a liquid which can readily be poured, to the solid commonly known at ordinary temperatures and which has a hardness comparable to that of the hardest steel. In spite of this hardness, a rod of glass supported at the ends will sag or "flow" under its own weight in the course of months or years.

Beilby has shown that amorphous metal is produced in the operation of polishing or rubbing done at room temperature. Such rubbing, however, can be produced not only by polishing the surface but by rubbing the crystals themselves over each other, or by block movements of crystals along certain crystallographic planes known as "gliding" planes or "slip" planes. This takes place, for example, in the cold drawing or cold rolling of metals. Such cold working results, according to Beilby, in the formation of amorphous metal films at the planes of rubbing, and produces hardening as layers of cementing material of a vitreous, amorphous character are formed at all surfaces of slip or shear throughout the mass. There is a limit to the amount of cold deformation that a metal will stand, and further attempts cause rupture. According to Beilby, the fact that overdrawing reduces the strength is due to the exhaustion of available crystalline material, and he regards the crystal modification of the metal as the sole source of plasticity. When the available supply of crystals is exhausted the metal becomes essentially glasslike in structure, and, like glass, incapable of further deformation.

Metals which have been hardened by cold work are softened by heating to a suitable temperature. The softening is accompanied by the restoration of ductility and by the substitution of a clearly crystalline structure of "equiaxed" grains (i.e., grains whose diameters are approximately the same in all directions) for the disordered structure found after cold working. This operation is called "annealing," and is a regular part of most industrial processes involving the cold deformation of metals. The softening is explained by Beilby's theory as due to the recrystallization of the amorphous metal produced by the cold work. Disappearance of the amorphous metal would, of course, involve the simultaneous disappearance of the hardness and brittleness supposed to be due to it. This would explain why in such operations as cold drawing it is necessary to anneal metal between passes.

Intercrystalline Cement

From what has been said above, it would appear that metal such as iron or steel consists of a jumble of grains, of which no two lying side by side are oriented in the same direction, with the irregular spaces between the grains (the black spots in Fig. 10) filled with an amorphous material. The remarkable part about this is that when pure metals at room temperature are broken, the fracture passes almost exclusively through the grains rather than between them, from which it would appear that the grains are weaker than the boundary material. When, however, metals are broken at temperature closely approaching their melting points, fracture follows the intergrain spaces rather than the grains themselves. Furthermore, permanent deformation of metals consists of plastic deformation of the crystalline grains by the process of slip. In the case of ductile metals, fracture through the grains is necessarily preceded by some plastic deformation of the grains and a general deformation of the metal as a whole, so that when the fracture takes place through the grain it is accompanied by a ductile break. All of these and some other facts lead to the belief that the grains of metals are held together by a cement of a vitreous, amorphous nature. This cement is hard and strong at the lower temperatures, stronger in fact than the grains. As the temperature rises the amorphous cement softens, and Bengough[5] has expressed the opinion that it disappears entirely at high temperatures, where the grains readily pull apart without any appreciable deformation.

Of late two factors have come to light which apparently support this theory. One is the result of an investigation by Dr. A. P. Davey in which it was found that single large grains of copper have a materially higher electrical conductivity than the purest copper wire (the electrical conductivity of granular is usually higher than that of amorphous materials). The other fact which may have a bearing on the same subject is the remarkable ability of high-chromium metals to retain their strength and especially hardness at high temperatures where ordinary steel is comparatively weak.

Beilby has also advanced the theory of hardening by the formation of amorphous metal, and Rosenhain expresses the belief that the great hardness of hardened steel is due to the presence of an extremely minute network of amorphous layers surrounding the very fine grains of alpha iron which result from the rapid transformation of gamma iron. This would explain why two such ap-

parently opposite operations as cold working on the one hand and quenching of hot metal in water or oil on the other, produce apparently similar results, and why annealing in both cases relieves the metal of its hardness.

RECRYSTALLIZATION

It has been stated above that severe cold working produces hardness in metals and that subsequent annealing removes this hardness. Further investigation has indicated that each metal has a temperature at which the grain structure distorted by cold working is replaced by new grains usually of approximately equiaxed shape (by "equiaxed shape" is meant one whose diameter is approximately the same in all directions). The change produced by heating to or above this temperature (which for iron is a minimum of 450 deg. cent.) when the strain-hardening effect is lost, is called "recrystallization." It is not intended to go into a full discussion of this subject here, and the reader is referred for further information to Chapter 5 of Jeffries and Archer's Science of Metals, from which the following paragraph, however, is taken.

> Iron is unique in its grain-growth characteristics because of the allotropic change in a temperature region permitting grain growth. If iron is cast, it forms grains of gamma iron above 900 deg. cent., crystallizing with a face-centered cubic lattice. On cooling, gamma iron changes completely into a body-centered lattice at about 900 deg. cent., and this transformation gives rise to entirely new grains. If the iron is kept below 900 deg. cent., it behaves much like any other metal within a similar temperature range. It can be cold-worked and hot-worked, and the grain-growth characteristics can be studied at temperatures up to 900 deg. cent. On heating above (about) 900 deg. cent., however, the structure again changes completely into new grains of gamma iron. Every time iron is caused to change from one crystal lattice to the other a new set of grains is produced. Pure gamma iron cannot be cold-worked, because at the lowest temperatures at which it is stable it cannot be strain-hardened.

It has been found also that the greater the degree of strain hardening, the lower will be the temperature of recrystallization. The recrystallization temperature, however, not only varies for different treatments for different pieces of the same metal, but it varies within the same piece of metal if a homogeneous condition does not exist. With mild cold deformation and large grains it is impossible to produce a condition even approximating homogeneity, and such samples usually recrystallize over a considerable range of temperature. The greater the amount of strain hardening, the smaller this range will be, i.e., the complete change in structure from strained to unstrained grains will take place within a smaller temperature range. Furthermore, the lower the recrystallization temperature, the smaller the grain size, which explains why overheating steel causes what is known as a "coarse" grain.

CRYSTALLINE STRUCTURE OF METALS, THEIR STRENGTH AND HARDNESS

It has been shown above that metals such as iron consist essentially of crystals having, when not deformed by their own interaction or outside agencies, the shape of cubes with atoms located in the corners and additional atoms in the center of the body or the center of each of the faces of the cubes, and while the distance between these atoms has to be measured in Ångstrom units, each of which is roughly one-two hundred and fifty millionth part of an inch, these distances are quite large as compared with the size of the atom itself. Furthermore, these distances are not accidental but are due to the action of forces holding the atoms together, and the atoms vigorously oppose every effort to change the interatomic distances within the crystal or the distances between the crystals. If an attempt is made to bring the atoms closer, as, for example, by applying compressive stresses to a metal, the repulsive forces between the atoms come into play and the amount of force necessary to overcome them expresses the compressive load. When an effort is made to pull a metal apart, the cohesive stresses oppose it.

It may be of interest in a general manner to see just what factors affect the character of the crystal and grain structure of metals. As has been already stated, rapid cooling from the liquid state tends to result in a fine grain structure, and slow cooling in extended dendrites and coarse grains. It has been observed, also, that the presence of impurities at times in very small quantities may affect the crystalline structure of iron. This may explain why the addition of such a small amount as 0.1 per cent of vanadium or molybdenum has so great an effect on the mechanical properties of steel and markedly increases the ability of the metal to resist alternating stresses. Attention has also been called to the fact that cold working followed by heating to a proper temperature produces a refinement and rearrangement of the grains. In a way hot working is equivalent to cold working and heating, with the distinction, however, that the working and heating are simultaneous instead of the heating following the working. It produces a refinement of the grain essentially in the same manner as heating after cold working, but also produces a tendency for the grains to rearrange themselves in the direction of working. This may be called a "pseudo-fibrous" condition of metal.

The hardness of solid solutions is due, according to Rosenhain, to distortion of the normal space lattice of one of the metals by the substitution of atoms of the other metal for its own atoms, this distortion serving as a hindrance to slip on the crystal planes. The more soluble in each other two metals are, the less distortion is produced; the less soluble, the more distortion and hence hardness of the resulting product. This explains why gold and silver alloys are soft (the two metals being soluble in all proportions), while carbon, being but slightly soluble in gamma iron, produces a hard material.

Jeffries and Archer have ascribed hardness of metals to what they call "slip interference" which means the presence of a structural condition which interferes with slip along the cleavage planes of the crystals of the metal. In a pure metal this would be produced by grain refinement in cold-worked metal, by the fragmentation of the grains; while the slip within the grains is opposed by the presence of a hard constituent uniformly distributed, the effect of the hard constituent depending on its amount and, within certain limits, the fineness of its subdivision.

PLASTIC STRAIN IN METALS

According to H. J. Gough,[8] the mechanism of plastic strain in metals is a process of slip on crystallographic planes. He bases this conclusion on the result of the work of Taylor and Elam.[9] The measurements carried out by these investigators on a single crystal of aluminum showed that for extensions up to and including 40 per cent, distortion occurred by a process of slip or shearing on one set of octahedral planes; further, the direction of the slip was along one of the three principal lines of atoms in this octahedral plane. At 40 per cent extension of the specimen a second set of octahedral planes assumed the same inclination to the axis of the specimen as that set of planes on which slip was taking place. Subsequent distortion occurred by simultaneous slip on both sets of planes. It was also brought out in this research that although the origin of the crystal axes with respect to the axis of the specimen was changing during the earlier stages of the tests, yet the crystalline symmetry was preserved at all stages of the test. In other words, measurement of the dihedral angles contained by the crystallographic planes showed that even in the last stages of distortion the specimen remained a single crystal.

It may be asked why the process of slip once started, due to the application of load, is arrested before fracture occurs. An explanation has been offered that slip in a body-centered cubic atomic arrangement is arrested by a transition to the hexagonal close-packed system which possesses only one set of planes along which easy gliding can occur. On the whole, as between Beilby and Rosenhain and their followers, two theories exist of the mechanism of plastic strain of a crystalline substance. First, that when the limiting elastic strain of the space lattice has been exceeded, gliding motion occurs by a process of slip along atomic planes; the amount of relative movement between two adjacent planes being a multiple of the interatomic distance, hence when such movement has ceased, the lattice distances and angles remain as before movement took place. Secondly, the amorphous theory requires that on the surface of the slip planes on which gliding has occurred the crystalline structure shall be completely broken down, resulting in the formation of non-crystalline material on these planes. A third theory was recently proposed to the effect that the mechanism of plastic flow, although at first characterized by the tendency to form gliding planes, nevertheless proceeds during the major portion of the flow period without any recognizable formation of these gliding planes. During this latter period the space lattice can be entirely destroyed, both lattice distances and lattice angles being permanently distorted.

The foregoing brief discussion of the relation between the plastic flow in metals and the crystalline structure of the latter has been introduced here largely to show what a vivid light the study of the atomic structure of metals can throw on such matters of interest to the mechanical engineer as the theory of elasticity and strength of metals. It should be remembered in this connection that the whole subject of investigating the crystalline structure of metals is extremely young—in fact, barely ten years old—and if so much information has already been obtained in that brief period it is easy to see how much more may be expected from this new method of investigation when its proper technique and the necessary apparatus have been developed.

Widmanstaetten Figures

What is known as "primary crystallization" occurs within the range of temperatures between the points *A* and *E* of the constitutional diagram of the iron-carbon system given on p. 345 of Mechanical Engineering, May, 1925, or between 1500 and about 1150 deg. cent. This crystallization leads to the formation of crystallites which are subsequently converted into dendrites. At lower temperatures what is known as "secondary crystallization" may take place. One of the forms resulting from this secondary crystallization is known as "Widmanstaetten figures." They occur if the interval range of secondary crystallization which lies between the point *G* or 900 deg. and some point below the line *PSQ* but near it, is covered too rapidly. A certain amount of the proeutectoid element separates then more or less instantaneously throughout every grain, this separation following the crystallographic planes of the octahedron.

Such a structure was first observed in 1808 on the meteorite, known as Agram Iron, by Alois Widmanstaetten (at that time director of the Imperial Porcelain Works at Vienna). Later on, somewhat similar structures were found here and there in ingots, and finally Col. N. T. Belaiew[5] produced an alloy in which the meteoric structure was artificially duplicated. Carl Benedicks[6] conceived the idea of suppressing the possible dendritic crystallization by first rapidly cooling from the melt (from about 3000 deg.), then bringing it again in the gamma zone (a possible granulation zone), and finally letting the alloy cool as slowly as possible through the area of secondary crystallization and the eutectoid zone.

His insight was rewarded by a brilliant success, as in his second melt he was able to obtain a large amount of iron-nickel constituents of the type found in meteorites, as well as a fine and distinct Widmanstaetten structure.

The following significant conclusion is drawn from this investigation by Belaiew: "The structure of meteoric irons is no more a mystery to us, but in solving that mystery the meteorites seemed to bring to us a message causing us to know and understand that the same laws govern the synthesis and crystallization of matter both here on our earth, and there in the vast infinite space whence the meteorites come."

How the Crystalline Structure of Metals Was Determined

It may be of interest to state briefly how all this knowledge about the crystal structure of metals has been collected. It is comparatively simple to investigate the crystalline structure of such materials as rock salt, because they are transparent, but iron is not. Some means had to be found to "see" the structure of the metal, and these means were found as a by-product of a truly theoretical investigation which at first did not promise to be of any interest to the metallurgist.

Some years ago a good deal of doubt was felt as to the true nature of X-rays, namely, whether they were corpuscles traveling at immensely high speeds or waves similar to light waves. In the case of ordinary light this question is determined by measuring the wave of light by means of a diffraction grating, but the X-ray wave is extremely short—from 1000 to 50,000 times shorter than the shortest light wave—and a diffraction grating to permit the measurement of even the largest X-ray wave would have to have some 20,000,000 lines to the inch. It is obviously impossible to make such a grating, even by the finest ruling machines that we know of.

Under these conditions Dr. von Laue had the brilliant idea of employing a natural crystal as a diffraction grating, his assumption, which he first developed mathematically, being that the atoms in a crystal ought to be regularly arranged, and that if a number of crystal units such as are present in an ordinary crystallite were taken, a sort of grating would be formed by the interatomic spacings throughout the crystal. While the method employed by Laue proved to be cumbrous, it confirmed his assumption and permitted measuring the lengths of waves constituting X-rays, incidentally proving that the only essential difference between visible light and X-rays is the length of their respective waves.

When the length of waves of X-rays of various kinds became known, several attempts were made, with ultimate success, to apply X-rays to the determination of the structure of various crystals. This was done by Sir W. H. Bragg and W. L. Bragg in England, Scherer and Debye in Germany, and Albert B. Hull and A. P. Davey in this country, Hull having also developed a modification of the Laue-Bragg process known as the "powder method" of crystal analysis. (For a brief description of the powder method, compare Jeffries and Archer, pp. 55–57.)

One of the remarkable features of the application of X-ray analysis and its modification to the investigation of crystalline structure of metals is the high precision of the results obtained. It has recently been stated by Dr. Walter Rosenhain in a lecture before the Society of Arts (February 16, 1925) in London, that the distances between the atoms in a great many crystalline substances are known to within an accuracy of much less than one-half of the one per cent. To understand the full significance of this statement, one has to realize that these measurements are effected in Ångstrom units, each of which is one-ten millionth part of a millimeter, or about one-two hundred and fifty millionth part of an inch.

Quite recently the General Electric Company, using an idea proposed by a German scientist, Dr. H. Barkhausen, developed a little device by means of which the noise produced by the movement of atoms when a rearrangement therein takes place can be plainly heard. This device consists of a cascade vacuum-tube amplifier of essentially the same design as used on some of the radio receiver outfits, connected to a pick-up coil having 17,000 turns. When an iron bar is inserted into this coil through which the current is flowing it becomes magnetized and this produces some kind of a shifting of the atoms, which is heard in the head phones of the amplifier as a roar. If the bar is rapidly withdrawn from the coil and reinserted, only a slight amount of the magnetization is lost, and accordingly when the bar is reinserted, instead of a roar there is only a slight noise. The device operates, of course, on a principle entirely different from another device (the "clink" detector for discovering fissures in forgings) which will be described later in this series of articles.

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Unilateral and Bilateral Tolerances as Applied to Interchangeable Manufacture[1]

A Discussion of the Two Tolerance Systems and Their Application to "Standard-Hole" and "Standard-Shaft" Practices

By H. W. BEARCE,[2] WASHINGTON, D. C.

DURING the past few years many technical articles[3] have been published in various engineering journals on the subject of unilateral vs. bilateral tolerances. These articles indicate that in general those who have gone into the question in detail have reached the conclusion that from a practical standpoint the unilateral system of tolerances has certain advantages over the bilateral.

There are, however, many competent gage men—inspectors, designers and engineers—who are more familiar with bilateral tolerances and who regard that system as preferable. There appears also to be a lack of complete understanding as to just what is meant by the terms "unilateral" and "bilateral" as applied to tolerances. There need be no mystery or uncertainty in this regard. Briefly defined, "unilateral" means "one-way" and "bilateral" means "two-way."

In what follows certain considerations with reference to tolerances will be discussed in detail, and in order that there may be no possibility of misunderstanding, certain terms will be defined as they will be used throughout this paper.

1 *Tolerance.* The amount of variation permitted in the size of a part. In the unilateral system it is the difference between maximum and minimum size. In the bilateral system it is the permissible variation from a nominal or specified size.

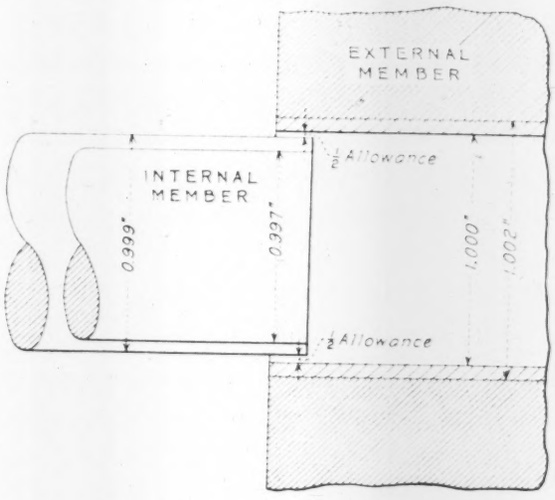

FIG. 1 ILLUSTRATING TOLERANCES AND ALLOWANCES

Tolerance: external member = + 0.002 in.
internal member = − 0.002 in.
Allowance: ..... = 0.001 in.

2 *Allowance.* An intentional difference in the dimensions of mating parts. It is the minimum clearance or the maximum interference between mating parts. It represents the condition of the tightest permissible fit, or the largest internal member mated with the smallest external member. It is to provide for different classes of fit.

The terms "minimum allowance" and "maximum allowance" are sometimes used to designate the tightest and loosest conditions of fit. This usage, however, is not in accordance with the above definition. These conditions are more properly designated "minimum clearance" and "maximum clearance," or "minimum interference" and "maximum interference," respectively.

3 *Unilateral Tolerances.* One-way tolerances. They are applied in one direction only to the *limiting* size of a part.

4 *Bilateral Tolerances.* Two-way tolerances. They are applied in both directions to the specified, or "ideal," size of a part. The specified size may or may not be the mean size. That is, the plus and the minus tolerances are not necessarily equal. For example, they might be equal, as 1.000 in. ±0.001 in., or unequal, as 1.000 in. +0.002 in.
−0.001 in.

5 *Basic Size.* The theoretical or nominal standard size from which all variations are made.

As an example to illustrate the above definitions, a plain plug and ring of 1 in. nominal size may be selected and the following conditions specified:

1 The allowance, or the difference between the smallest ring and the largest plug, shall be 0.001 in.

2 The tolerance on the ring shall be, under the unilateral system, 0.002 in.; under the bilateral system, ±0.001 in.

3 The tolerance on the plug shall be, under the unilateral system, 0.002 in.; under the bilateral system, ±0.001 in.

While no condition has been imposed as to the actual limiting sizes of either the plug or the ring, the above conditions fix the maximum tightness and the maximum looseness that will be permitted.

The specified conditions can now be completely expressed in terms of either unilateral or bilateral tolerances.

Under the unilateral system the dimensions, allowances, and tolerances will be as follows:

DIMENSIONS OF RING

Basic Size	Tolerance	Specified on Drawing	Maximum	Minimum
1.000 in.	+0.002 in. −0.000 in.	1.000 in. +0.002 in. −0.000 in.	1.002 in.	1.000 in.

DIMENSIONS OF PLUG

Basic Size	Allowance	Tolerance	Specified on Drawing	Maximum	Minimum
1.000 in.	0.001 in.	+0.000 in. −0.002 in.	0.999 in. +0.000 in. −0.002 in.	0.999 in.	0.997 in.

Under the bilateral system of tolerances the same character of fit may be expressed thus:

DIMENSIONS OF RING

Mean Size	Tolerance	Specified on Drawing	Maximum	Minimum
1.000 in.	±0.001 in.	1.000 in. ±0.001 in.	1.001 in.	0.999 in.

DIMENSIONS OF PLUG

Mean Size	Allowance	Tolerance	Specified on Drawing	Maximum	Minimum
0.997 in.	0.001 in.	±0.001 in.	0.997 in. ±0.001 in.	0.998 in.	0.996 in.

It will be noticed that while the same character of fit is obtained under the two systems of tolerances as carried out in the foregoing examples, the limiting dimensions differ slightly. This is not necessarily the case. That is, under the bilateral system it would be possible to select 1.001 in. as the mean size of the ring. The limiting sizes would then agree with those obtained under the unilateral system, and would be expressed as follows:

[1] Published by permission of the Director of the Bureau of Standards.

[2] Bureau of Standards.

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Dimensions of Ring

Mean Size	Tolerance	Specified on Drawing	Maximum	Minimum
1.001 in.	±0.001 in.	1.001 in. ±0.001 in.	1.002 in.	1.000 in.

Dimensions of Plug

Mean Size	Allowance	Tolerance	Specified on Drawing	Maximum	Minimum
0.998 in.	0.001 in.	±0.001 in.	0.998 in. ±0.001 in.	0.999 in.	0.997 in.

It is possible, under either system, to change the tolerances, the allowances, or the character of fit in a definite, predetermined manner.

Suppose, for example, that it is desired to increase the tolerance on one or both of the members in order to reduce the cost of production, and at the same time to insure that the increased tolerance will always be in the direction of greater looseness; the tightest condition remaining unchanged. Obviously the thing to do is to keep the allowance unchanged, and to make such changes in the tolerances as may be required to bring about the desired result.

In the foregoing example let it be agreed that the tolerance on each member shall be increased by 50 per cent.

Under the unilateral system the tolerance will become 0.003 in., and the limiting dimensions will be as follows:

Dimensions of Ring

Basic Size	Tolerance	Specified on Drawing	Maximum	Minimum
1.000 in.	+0.003 in. −0.000 in.	1.000 in. +0.003 in. −0.000 in.	1.003 in.	1.000 in.

Dimensions of Plug

Basic Size	Allowance	Tolerance	Specified on Drawing	Maximum	Minimum
1.000 in.	0.001 in.	+0.000 in. −0.003 in.	0.999 in. +0.000 in. −0.003 in.	0.999 in.	0.996 in.

Under the bilateral system the same thing may be accomplished. It may be done by a straight increase of 50 per cent in the tolerance on both members, as in the case of the unilateral tolerance. This change, however, will require a change in the mean size of one or both members in order that the tightest condition shall remain unchanged.

If the mean size of the ring be kept at 1.000 in. as before, the tolerances and other dimensions, under the bilateral system, will be as follows:

Dimensions of Ring

Mean Size	Tolerance	Specified on Drawing	Maximum	Minimum
1.000 in.	±0.0015 in.	1.000 in. ±0.0015 in.	1.0015 in.	0.9985 in.

Dimensions of Plug

Mean Size	Allowance	Tolerance	Specified on Drawing	Maximum	Minimum
0.996 in.	0.001 in.	±0.0015 in.	0.996 in. ±0.0015 in.	0.9975 in.	0.9945 in.

It will be be seen that the above change of tolerance has resulted in a change of all limiting sizes of both the plug and the ring, and therefore of all gages, both "go" and "not go."

It would be possible, however, under the bilateral system, to keep the specified size, the minimum ring, the maximum plug, and the allowance all unchanged, and to put all of the increased tolerance in one direction from the specified size, which will no longer be the mean size. Under such an arrangement the dimensions become:

Dimensions of Ring

Specified Size	Tolerance	Specified on Drawing	Maximum	Minumum
1.000 in.	+0.002 in. −0.001 in.	1.000 in. +0.002 in. −0.001 in.	1.002 in.	0.999 in.

Dimensions of Plug

Specified Size	Allowance	Tolerance	Specified on Drawing	Maximum	Minimum
0.997 in.	0.001 in.	+0.001 in. −0.002 in.	0.997 in. +0.001 in. −0.002 in.	0.998 in.	0.995 in.

It will be seen from the above that under the bilateral system a change of tolerance, in one direction from the specified size, may be so made as to change only the "not go" or the "go" gages, as desired, in the same manner as under the unilateral system.

It is apparent from the foregoing that if certain conditions are specified these conditions may be expressed in either unilateral or bilateral terms. Up to this point, then, the terms "unilateral" and "bilateral" simply designate a method of expression rather than a fundamental principle of design. The unilateral system, however, usually carries with it another important principle, namely, that of keeping constant the extreme size of one member. This principle is followed out in two ways known, respectively, as "Standard-Hole Practice" and "Standard-Shaft Practice." It is this combination of unilateral tolerances with standard-hole practice or, in special cases, standard-shaft practice, that makes for maximum simplicity of design and economy of production.

On the other hand the bilateral system carries with it a "mean size," "specified size," or "ideal size," from which variations in both directions are made. This imaginary or ideal size combined with variations in both directions tends to make the bilateral system confusing and difficult to put into effect.

The claim is sometimes made by advocates of the bilateral system of tolerances that the use of this system results in the production of parts which, on the average, are closer to the "ideal size" than are similar parts produced under the unilateral system. This claim is based on two assumptions: first, that under the bilateral system the specified size is the ideal or best size, and second that this ideal or best size will, on the average, be more nearly attained under a bilateral than under a unilateral system. Both of these assumptions are open to question.

In the first place, what ground is there for the assumption that the mean size, or some other specified size between the permissible limits, is better than any other size within those limits? In some cases a size near the mean is perhaps to be preferred, while in other cases a size as close as possible to the maximum metal, that is, the largest permissible shaft, in the smallest permissible hole, is closest to the ideal, since it gives the greatest possible wear and hence the longest life.

The second assumption, namely, that the naming of the mean size or other "ideal" size will result in the production of parts closer to that size than will the naming of the limits, is open to question. If all parts were "measured" individually as they were being produced, then the naming of the desired or ideal size would doubtless tend to make that size serve as a target about which would be grouped the sizes of the parts produced. But in the actual production on an interchangeable basis the parts are not measured individually as they are being produced but are gaged for conformity to the predetermined limits. The result is that in general the mean size, or any other specified size between the limits, is of little or no importance in interchangeable manufacture. The manufacturer and the inspector are concerned only with the question of whether or not each part produced is within the prescribed limits, since any part which is within these limits is acceptable. If any such accepted part is unsatisfactory, then either the limits have been improperly set or the method of inspection is inadequate.

Advantages of Standard-Hole Practice

The advantages to be derived from the use of a uniform minimum hole for all fits, usually designated as "standard-hole practice," are those which naturally follow from keeping constant those members which can most easily be kept constant and varying those members which can most easily be varied. These advantages are:

1 A single set of fixed tools for all fits

2 A single set of "go" plug gages for all fits.

Since, in general, holes are made with fixed tools such as drills, reamers, and taps, while internal members, shafts, bolts, etc. are usually made with tools which are readily adjustable as to size, such as dies, chasers, grinding machines, etc. it is apparent that the internal member can, in general, be more readily varied in size than can the external member. It is therefore good practice to take advantage of the difference in methods and tools employed in producing external and internal members, and to make all necessary variation in the internal member when various degrees of tightness of fit are required, keeping the external member constant for all fits.

However, when conditions are such that the advantages ordinarily

resulting from the standard-hole practice do not exist, or are more than offset by the difficulties introduced by its use, it should not be followed. That there are such cases will be readily admitted even by the most ardent advocates of the standard-hole practice. A good example of such a case is that of a machine tool in which several kinds of fit are required on the same shaft. It would be folly to advocate the use of a shaft of several different diameters in order to produce the desired fits, when the fits could be much more economically obtained by using a standard shaft and varying the holes as required. In this case the holes would probably be ground, and could be as easily varied as could the shaft. Such cases, however, should be regarded as exceptions to which the usual advantages of standard-hole practice do not apply, rather than as indicating that the practice is without merit.

In this as in other questions of design and production, general practice should be governed by ordinary conditions, and exceptional cases by the special conditions to be met. As a general practice for ordinary conditions of interchangeable manufacture the standard-hole practice tends toward economical production.

As already pointed out, unilateral tolerances are applicable to a system of dimensions based on either standard-hole practice or standard-shaft practice.

After the correct limiting sizes of parts to give the character of fit desired have once been arrived at, it is of little importance whether unilateral or bilateral tolerances are used in maintaining these sizes.

In the example already given it has been shown that the tolerance on one or both members can be changed under either system without changing the minimum hole or the maximum plug. Under the bilateral system, however, the change of tolerances would necessarily involve either a change in the specified "ideal size" or the introduction of unequal tolerances in the two directions from the specified size.

Under a unilateral system of tolerances all that is necessary in order to put into effect any desired change of this character is to change the high limit of the ring and the low limit of the plug. This involves changing only the "not go" gages.

In the example given it was desired to change the tolerance on each part from 0.002 in. to 0.003 in. This was accomplished simply by changing the maximum ring from 1.002 in. to 1.003 in., and the minimum plug from 0.997 in. to 0.996 in.

Another important point in favor of unilateral tolerances, which has been implied but not definitely stated, is that with this system "standard" or "basic-size" gages can be used to a maximum extent. For example, the use of basic-size "go" gages will insure that any products which pass a mating plug and ring will assemble at least as freely as the plug and ring with which they are inspected.

If it is desired to provide a minimum looseness somewhat looser than a metal-to-metal fit, then it will be necessary either to decrease the maximum size of the internal member (shaft), or to increase the minimum size of the external member (hole). The standard or basic gage will, however, still serve as the limiting size for that member which is retained as basic.

With standard-hole practice and unilateral tolerances a standard "go" plug of basic size will serve for all classes of fits. This means a considerable saving in the cost of gages.

SPECIFIED LIMITS SHOULD NOT BE EXCEEDED

There is a fundamental principle in gaging which has not yet been mentioned and which should not be overlooked. That is, that the specified limits of the product represent the extremes which should not be exceeded as a result of gage tolerance, gage wear, or from any other cause. These limiting sizes represent the most extreme conditions of tightness and looseness which the designer is willing to permit.

In order that this principle may be carried into effect it is apparent that all gage tolerances must be so placed, with reference to extreme size of part, as to insure always that the part in question does not exceed the specified limits. From this condition it follows that a gage which represents a minimum limit may be larger but never smaller than the minimum size specified for the part, while a gage which represents a maximum limit may be smaller but never larger than the maximum size specified for the part. That is, all gage tolerances must be in such a direction as to tend to bring the "go" and "not go" gages nearer together. This principle applies to both unilateral and bilateral systems.

While the application of the above principle will always result in keeping the accepted product within the specified limits, it is apparent that it will also result in decreasing the tolerance available to the manufacturer, by whatever amount is used on the gages themselves.

The maintaining of a proper balance between product tolerance and gage tolerance is a matter of the highest importance, since upon it, in large measure, depend the cost and effectiveness of gaging.

On this point the author must take exception to the practice which was advocated by Sir Richard Glazebrook in his Thomas Hawksley Lecture before the Institution of Mechanical Engineers, on November 5, 1920, and recently reaffirmed by the British Engineering Standards Association in their British Standard Limits and Fits for Engineering (No. 164, 1924): namely, that of placing the tolerance on inspection gages outside the product limits.

The prime object of gaging is to insure that no accepted product is outside the predetermined limits of acceptance. There can, therefore, be no justification for setting up an elaborate system of gaging only to violate and vitiate it in the final inspection by the acceptance of parts which may be outside these limits.

Any parts which are so close to either rejection limit as to be improperly rejected as a result of tolerance or wear of the inspection gage, should be reinspected with a gage as close to the product limit as is practicable, but with all errors in the direction of safety rather than in the direction of danger.

Whether unilateral or bilateral tolerances are used, the product limit should in all cases be regarded as a "dead line" beyond which no parts should be accepted as the result of gage error, gage wear, or any other cause. Only by strict adherence to this fundamental principle can limit gaging attain its maximum value in interchangeable manufacture.

Fuel Oil in the Commercial Buildings of New York

BY A. M. COYLE,[1] NEW YORK, N. Y.

THE authorities of New York City no longer regard fuel oil as a dangerous explosive. The question now is not whether we can get approvals for the installation of oil-burning plants, but whether we can get oil at a price which will make future oil burning economical.

Experiments in the use of oil as fuel under steam boilers began in this country about forty years ago. The oil used in these experiments was of a grade which could be sprayed without heating. Spraying was effected by steam or compressed air. The spray was usually projected against a plate to cause complete combustion. Later the oil was sprayed over checker plates. These devices were necessary because the early sprays were not effective and did not spread the flame so as to bring about the proper mixture of air. Checker plates are still in use to some extent, but are rapidly being superseded by burners which produce conical sprays.

Before fuel oil began to be used in the commercial buildings of New York City, it had come into extensive use elsewhere. The problems attending its introduction here were therefore not problems of burning but of storage. Prior to 1919, oil had been burned in the city limits in a number of industrial plants, for annealing and similar purposes, but up to that time the authorities had established no code of rules governing the storage and burning of fuel oil. The

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storage of oil in or near buildings had been steadily disapproved by the Fire Underwriters, the Bureau of Fire Prevention, and the Fire Department. When, therefore, in 1919 there developed an insistent demand for the right to install oil-burning systems in the city, it became necessary to establish rules to govern these installations. The discussion regarding such rules focused about the plant being installed in the Singer Building. We have to give credit to the men who installed this plant for the patience and tact with which they carried out a difficult undertaking, and met and overcame bureaucratic objections. While there is still some fear of fuel-oil installation in fireproof buildings, and while changes are still desirable in regulations, the phase of conflict and irritation has passed.

The records of the Bureau of Fire Prevention show that they have inspected and passed upon between three and four hundred oil-burning installations. Two-thirds of these are small equipments, using one or two burners, but the remainder are those in large office buildings, hospitals, department stores, hotels, loft buildings, theaters, banks, manufacturing plants, and apartment houses. The small installations, of course, use light oils which do not require preheating.

Classification of Oil-Burning Methods

Burners for fuel oil may be classified as follows:

1 Burners for Domestic Heating or Use in Small Buildings. In these light oil is used which is vaporized either by a small fan passing air through a jet or by passing the oil over a small plate heated by a gas flame. The flame also ignites the vapor which is mixed with the air delivered by the fan.

2 The Mechanical Spray Used in Industrial Plants and Under Small Boilers. This burner has a mechanism which is rotated either by a small motor or by the air from the fan. The oil is introduced into a cup and this creates a whirling motion. This is a very satisfactory burner for industrial use and it may be used for heating boilers, because the time may come when there will be no waste heat to create stack draft. A fan will then be necessary.

3 Burners Using Steam Atomization. Until quite recently steam atomization has been standard. The United States Navy after exhaustive tests decided in favor of mechanical atomization. This has given prestige to that system. The Navy's decision was based to some extent on the desirability of saving the water used in steam atomization.

4 Burners Using Mechanical Atomization. In all these burners the object is to produce a whirling spray. This is accomplished by internal passages tangential to the central chamber. In some cases the flow may be regulated within limits by varying the oil pressure. The other burners provide an adjusting valve within the burner. One company claims that by means of the overflow passage a range of adjustment may be obtained which is wider than that obtainable by other constructions.

Rating of Boilers

Several years ago the American Society of Heating and Ventilating Engineers put forward the following formula for calculating rating:

$$\text{Boiler capacity per hour in B.t.u.} = \frac{C \times H}{T}$$

where

C = pounds of dry fuel consumed
T = duration of firing period
H = total B.t.u. available at the boiler outlet per lb. of dry fuel

the length of firing period T to depend upon the size of boiler, kind of fuel and purpose of test.

This formula merely establishes the size of the firebox; it has nothing to do with the efficiency or capacity of the boiler. However, many manufacturers of small cast-iron boilers are taking advantage of it and are turning out boilers which are chiefly firebox and smoke flue. They give the impression that these boilers have the stamp of approval of the engineering societies. The construction results in very high stack temperature. Introduction of oil aggravates this situation very much because small burners of the type described do not touch the sides of the firebox and therefore they have very little heating surface above the firebox, which has to do all the steaming.

It is a great mistake to allow the installation of oil burners in boilers of this character. If we are going to use oil burners, we should have boilers that make use of the heat before it leaves the stack.

Some Typical Installations and Results

The Singer Building. The change over from coal to oil was made in 1919. The boilers are B. & W., and burners were installed by the same company. The furnaces were originally hand-fired. The grates were removed and burners installed with but slight alteration to the original setting. The combustion space is of course not as large as would be provided in present-day practice. This limits the extent to which the boilers can be forced. The boiler capacity of this plant is ample. There is no reason for forcing.

Since the plant was changed over to oil very careful records have been kept of the cost of fuel, operation, and maintenance. The saving effected by the use of oil has been estimated by considering the tonnage of coal used in previous years and figuring the cost of that tonnage at the current market price. The showing is very markedly in favor of oil.

It should be noted that since 1919 certain changes have been made in the plant. These changes consist in providing electric drives for various engine-room auxiliaries, pumps, air compressors, and refrigerating machines. It so happened that the electric units already in place were capable of taking on the added load of the auxiliaries to good advantage. In fact, the overall efficiency of these units was improved by the added load. After the change less steam per kilowatt was used than before. Another matter should be taken into consideration before accepting the records of this plant as a criterion of oil-burning economy: when coal was used the furnaces were hand-fired.

During the past five years the fuel market has been abnormal. The inactivity of shipping and the closing down of industries left an immense quantity of fuel oil available for urban boiler plants. In 1923 a number of unusually large oil pools were tapped. An overproduction followed and prices dropped. While these things were happening, the coal situation was at its worst. From now on we may expect stabilization and a downward tendency of coal prices and an upward tendency in the price of oil. Even allowing for all these considerations, the results at the Singer Building are favorable to oil. This year may not show any great saving in the cost of fuel and the next year may prove that oil is as costly as coal. Nevertheless the operating engineer at the Singer Building would not go back to coal unless actually forced to do so.

The R. H. Macy & Co. Department Store. This is the largest independent power plant in New York City. The electrical output is about 30,000 kw-hr. per day, and in addition steam is supplied for numerous auxiliaries and for cooking.

Oil was at first installed because the boilers when coal-fired were insufficient in capacity to take care of the load due to the new building. Additional boilers have been provided so that the boiler plant is now of ample capacity.

Owing to the many changes which have been made in this plant it has been impossible to analyze operating costs. The author has compared the records of electrical output and fuel bills for the years 1922 and 1924. The figures for the month of February in these two years are sufficient for illustration.

	Feb., 1922	Feb., 1924
Electrical output	302,800 kw-hr.	514,400 kw-hr.
Fuel bill	$8,984 (coal)	$8,658 (oil)

In the month of February, 1922, the price of coal was very high, being $6.75 a ton. The price of oil in 1924 was comparatively low, 4½ cents a gallon. When the boilers were coal-fired, coal was man-handled and used under inconvenient circumstances. Also since 1922 there have been a number of steam-driven auxiliaries changed over to electric drive, which has meant an economy, so that perhaps these figures are not as rosy as they look. This year will show a different result. The price of oil has gone up to 4¾ cents. The price of coal is considerably less. The only anxiety of the operating engineers at the Macy Building is to be able to get fuel oil at anything like the present price.

The Metropolitan Life Insurance Building is another first-class plant in which oil burning has been introduced. Four boilers are equipped for burning oil. Steam atomization over checker plates is used. In order to provide combustion space, Dutch ovens have been built in front of the boilers. The operation of this installation is satisfactory and economical. One thing which had weight in deciding the owners to burn oil was the question of ash removal. In 1922, when the streets were badly blocked with ice, ash removal from this building was seen to be a serious problem.

Oil burners have been installed in several important new buildings, notably the new Equitable and the Saks Fifth Avenue Store. In both Bethlehem Dahl burners and the Merritt control system are used. The burners in the Equitable have been in use only a short time; those in the Saks Store have been in use long enough to demonstrate that they are satisfactory. In the latter building steam is used for heating only. All power is supplied by the New York Edison Company.

The author believes that in plants using 300 kw. or more and in cases where exhaust steam can be put to good use, it is still possible to generate current cheaper than it can be purchased. How long this will continue to be the case is a question. Certainly the tendency is to the general use of public-service power. There are a good many things which accelerate this tendency: good salesmanship on the part of representatives of the power companies, the uncertainty of fuel supply, unsatisfactory labor, and a desire on the part of executives to be relieved of responsibilities outside of the main line of their commercial enterprise. Greater than all these are the items of first cost and space requirements, and a feeling that the whole subject of power development is in a state of flux. An installation which is up to date today may be obsolete long before it is worn out. Men who are engaged in commercial lines, the men who erect large city buildings, would rather have a definite contract for power than take chances on the installation of a power plant.

Some Questions at Issue in Oil Burning

We may expect in the future that fuel will be used in commercial buildings largely, if not exclusively, for the purpose of heating, and that brings up several questions which are of particular interest to heating and ventilating engineers.

In the first place, most of these heating plants use low-pressure steam. There is only a small difference in actual thermal value between low-pressure and high-pressure steam, but it is doubtful whether it pays to generate steam at high pressure and then reduce it to four or five pounds for heating. High-pressure steam is needed for bringing the oil to the proper temperature for spraying. It would seem advisable if mechanical burners are used, that a separate high-pressure unit be installed in all cases and the boilers used for heating purposes.

Another point should also be called to the attention of heating and ventilating engineers. In power plants there is now a great demand for the recovery of the heat lost from the stack case. The author does not know of any plant in New York where there is any attempt made to do this. It is all recoverable for heating purposes, because we can install stack radiators and recover that heat for use in the building, by use of a fan, and it is very available for ventilating purposes. That may bring us down to a point where we shall have to have forced draft.

The Prospect of Fuel Supply

Oil at present represents a very small fraction of the city's fuel. The oil companies are already hedging as to future prices and future deliveries. When the B.t.u. price of oil is less than the B.t.u. price of coal in New York, it is safe to say that the market is abnormal. If we do not actually experience oil shortage, we may certainly expect a time when the overall cost of oil firing will equal the overall cost of coal firing. Even then, many users will prefer oil.

The question of our vanishing oil supply is one about which there has been a great deal of discussion. The supply of oil is undoubtedly more uncertain than the supply of coal, and we really have no positive information on which to base an estimate. Refiners are all working toward an increased production of gasoline. This means a lessened supply of fuel oil. One refiner has even put forth the statement that within five years crude oil will be cracked so as to yield only gasoline and coal tar. Nevertheless the advantages of liquid fuel have been demonstrated and the demand will be supplied. Synthetic liquid fuel is a reasonable possibility, and another problem for the chemists is to find a method of increasing the hydrogen in heavy hydrocarbons or even of hydrogenating solid carbon.

At present there is being produced in the United States a little over one barrel of oil for every ton of coal mined. This does not mean that we have anything like that amount of fuel oil available. Not more than 20 per cent of the total is available for fuel. It is therefore out of the question to talk about oil displacing coal completely.

Under present conditions it is certainly necessary to provide for coal firing in case the oil supply is cut off, or in case the price of oil becomes prohibitive. In a great many cases this item is not given proper attention. The only provision made for burning coal is to take out the burners and install grates, an operation which requires some labor and takes time. After this the furnaces have to be hand-fired through the burner openings. This arrangement is unsatisfactory and is only tolerable as an emergency makeshift.

While one body of engineers have been workng on fuel-oil burning, another body have been developing equipment for burning pulverized coal. It is interesting to note that furnaces suitable for fuel oil are also suitable for pulverized coal, and that a change over can be made without alteration and firing continued with the same boiler-room force employed to manage the oil.

The author has recently inspected a pulverized-fuel installation which is reduced to extreme simplicity. It consists of a hopper, with table feed which delivers coal to a small tube mill. This mill is air-swept. The air used is preheated by being drawn around the stack. After passing through the mill, the air carrying the pulverized coal is delivered through tubes to the burners. The air pressure at the burners is about 1 in. water column. Coal carrying a considerable quantity of water may be handled. The burning is accomplished without noise.

Dicussion

In the discussion which followed, Kingsley L. Martin,[1] chairman of the meeting, cited the Giant Power plan of the State of Pennsylvania, where coal would be distilled before burning in a few enormous central stations. Coal tar and gasoline would be thus obtained in large quantities. Discussing the high price of oil, he thought it was due to the fact that Mexican oil was used for high-pressure boiler work, and that such oil was subject to varying taxation by the Mexican Government. Discussing the preheating of oil, a necessity before it could be burned, Chairman Martin recommended that an independent high-pressure boiler be used to heat it. With electric preheating he thought the expense would be abnormal.

R. P. Bolton[2] referred to a plant which he was managing on Fifth Avenue. The oil-burning system had cost $18,500 and the plant had to meet an annual contract for 20,000 barrels of oil at 4¾ cents a gallon while coal could be bought at $4.61 a ton. Mr Bolton stated that while oil burning looked very attractive, both preheating it and pumping it took power or fuel, and the net result might be less satisfactory than expected. He cited the case of a company which had changed twenty boilers from coal to oil by pulling down the bridgewall and firing the boilers from the rear end. Initially there were economies, but as their contract for oil ran out and the price of coal went down they had to return to coal, burning pulverized coal. Mr. Bolton also said that the opportunities for getting a stable supply of oil were diminishing. As to the domestic use of oil, he called attention to the skill required to operate the furnace and the difficulty of installing oil tanks and plants in residences. He further pointed out that the elimination of every high-pressure element was necessary for real economy in the plant. In hotels, apartment houses, etc., when changing from coal to oil, boilers were now being made to operate at heating pressure only. As advantages occurring from oil burning he mentioned the saving of space, which was sometimes important, and such benefits as doing away with ashes.

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General Properties of Ball Bearings

Fundamental Factors in Ball-Bearing Design—Load, Life, Fatigue, and Friction Torque

By HAAKON STYRI,[1] PHILADELPHIA, PA.

THE early development of ball bearings in machine construction about fifty years ago was more or less haphazard. A great number of designs were proposed and tried out, with the usual result that the simple ones survived.

Professor Stribeck on behalf of Deutsche Waffen und Munitions Fabriken (DWF), about 1900, was the first to thoroughly analyze ball bearings and to develop sound principles for their construction.

The late Henry Hess presented a very thorough report on Professor Stribeck's investigation before the A.S.M.E. in 1907.[2] In later years Professor Goodman in England, A. Palmgren (SKF), Sweden, Ahrens and Huber in Germany, and recently the late Mr. Macaulay (Skefko), England, have further discussed the theory and properties of ball bearings.

In what follows an effort is made to point out some of the fundamental factors in ball-bearing design and their application, and in this connection it will be of advantage to review briefly some of the results of Stribeck's investigations in reference to Hertz's work on the contact of elastic bodies.

According to Hertz, the amount of compression or the degree of approach of two bodies under load is proportional to $P^{2/3}$, and the diameter of the contact circle or the axis of the contact ellipse is proportional to $P^{1/3}$.

The dimensions of contact areas can easily be measured by etching under load or by making a previous copper etching and by measuring the imprint after pressure. This method is often used to determine the load imposed on a bearing, as it has been found that Hertz's theory is sufficiently accurate in this respect.

Professor Stribeck measured the deformation between balls and between balls and flat or curved surfaces under pressure, and the results obtained are in fair agreement with Hertz's formulas. It is worthy of note that there is no definite low-load limit where the permanent deformation disappears.

From these experiments as well as from theoretical calculations according to Hertz, we have the generally accepted statement that the load-carrying capacity (P) of a ball is proportional to the square of its diameter (d), or $P = kd^2$, where k is a factor called the "specific load," and usually expressed in kg. per $(1/8$ in.$)^2$.

Before discussing the permissible specific load, it will be well to consider how the loads are distributed in a ball bearing. In a thrust bearing the question is easy to answer, as the load on the bearing must be equally distributed on all the balls.

In a radial bearing, however, it is different. A load on the races is distributed radially through balls in the loaded half, provided the races have "line-to-line" fit, that is, a normal fit or one with no looseness nor tightness. If k is the specific load carried by the heaviest-loaded ball in the direction of the applied load, Stribeck has shown that the load on the bearing can be represented by $P = 0.2\ nkd^2$, where n = number of balls. If the bearing has a loose fit, a smaller number of balls will carry the load, and consequently the load P will be smaller for the same value k. If the bearing is tight, more balls will help in carrying the load, but excessive tightness may subject all the balls to the specific load k without applying any external load. (See Fig. 1.)

The influence of looseness and tightness in a radial bearing when applied as a thrust carrier should also be noted. In a spherical bearing the relation is simple, because there is practically no change in the contact angle for variation in looseness of the bearing, and the ball load under a given thrust remains constant. Not so for the deep-groove bearing, where evidently the looseness has a great effect on the contact angle. The groove radius also plays an important part, as seen in Fig. 2. Greater looseness and smaller groove radius will increase the initial contact angle and consequently reduce the ball load for any given thrust load.

It is worth while to study this condition a little more in detail.

It is known that for a definite pressure on a ball there is a corresponding definite deformation. If a line-to-line-fit bearing of normal curvature be considered, and an increasing thrust load be applied, the resultant load through the balls must first increase very fast and thereafter slower, because the contact angle increases due to the deformation. If a definite specific load on a ball is fixed as a limit, the corresponding load on the bearing will depend on the groove radius and the looseness, as well as on the ball size and the number of balls. This applies to so-called "radio-thrust" bearings as well as to the conventional annular bearings.

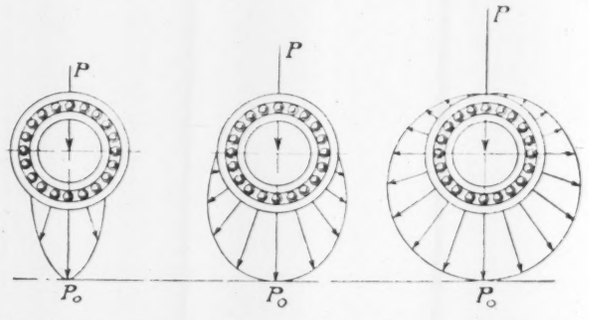

FIG. 1 LOAD DISTRIBUTION IN RADIAL BEARING (MACAULEY)
(P_o = maximum ball load; P = external bearing load.)

So far only bearing loads and ball loads under static conditions have been considered.

When rotating, the difference between a bearing taking radial load and one taking thrust load is more pronounced. If a radial bearing has a normal fit (line-to-line) after assembling on an arbor, it is easy to see that only one-half of the circumference of any of the races is subjected to load, and further, that the load will vary from 0 to the maximum ball load. (Fig. 1.) During rotation any point of the inner race will receive an impact when it passes a ball, and the magnitude of this impact will depend on the instantaneous position of the ball. In a bearing taking thrust load only, all of the impacts will be alike. The total number of impacts for a point on a rotating inner race can be expressed by $i = n\left(1 + \frac{d}{D}\right)$, where n = number of balls, d = ball diameter, and D = pitch diameter. These impacts must have a fatigue action.

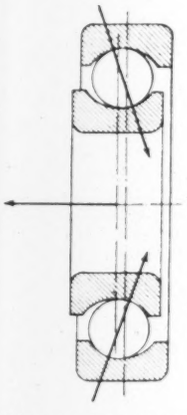

FIG. 2 BALL LOAD IN ANNULAR BEARING UNDER THRUST LOAD

In any material subjected to alternating load, it is known that there is usually some relation between the magnitude of the applied stress and the number of times the stress can be reversed before the material will fail from fatigue. For alternating stresses on the softer steels there is evidence of a definite fatigue limit, so that material stressed below this limit will have infinite endurance. It has been the author's concern to find whether there is such a fatigue limit for ball bearings, or to find how the average life would vary with the load.

When this problem was started on it seemed most logical to select a given size of bearing and a definite load, and repeat the test a sufficient number of times to find the probable variation in life as well as the average. Graphically the tests are shown in step diagrams, one step for each bearing, the ordinate giving the life in revolutions (Fig. 3). It is of interest to note how much the life varies for the same load conditions, and this holds for any type of bearing that has been tested. Similar curves for the life of

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[1] Chief, SKF Research Laboratory.

[2] Trans. A.S.M.E., vol. 29 (1907), p. 367.

Presented at a meeting of the Philadelphia Local Section of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, February 24, 1925.

human beings or of incandescent lamps show the same relation.[1] In Fig. 3 the arithmetical-average life corresponds to about 60 per cent of the failed bearings.

The next step in the investigation was to determine the influence of velocity of rotation. It was found that if life was determined in total number of revolutions, the velocity of revolutions would have no appreciable influence on the average life, at least up to 2000 r.p.m.

More important still was it to determine the variation of the

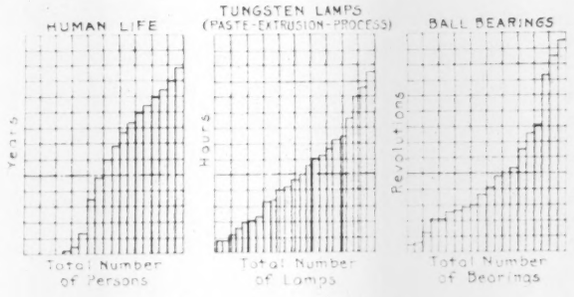


FIG. 3 LIFE CURVES (HEINDLHOFER AND SJÖVALL)

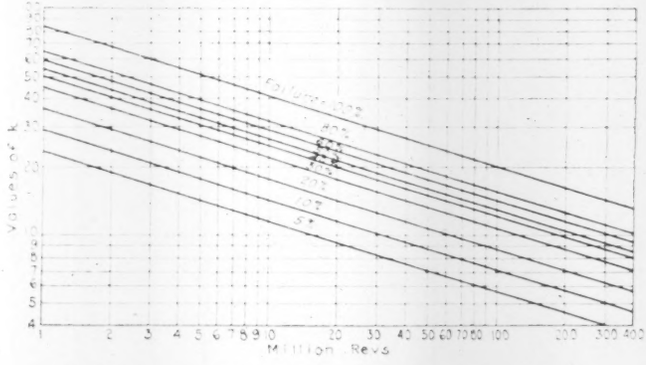


FIG. 4 BALL BEARINGS: LOAD-LIFE RELATION

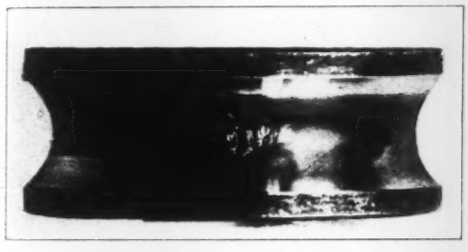


FIG. 5 CHARACTERISTIC FATIGUE FAILURE OF A BALL-BEARING RACE

average life with varying load, and for this test a great number of bearings from one lot were selected and run under four loads.

The results are shown in Fig. 4. On a logarithmic chart a practically straight line is obtained for the average life. For a decrease in load to 50 per cent the life is increased 10 times. Expressed in a formula, $k = CN^{0.3}$, where N = number of revolutions and C = a constant.

The slope of this line for average failures has been found to be practically the same for other types of bearings. Parallel lines can be plotted which represent various percentages of failure.

The curve has been determined for deep-groove bearings down to a specific load of 10 kg. per $(1/8$ in.$)^2$, and it took over two years to get the final results and the average at this load. The average for still lower loads will evidently consume a very much longer time, but a determination would be of great interest in showing whether or not a fatigue limit exists for ball bearings.

In the course of the work it was found above that in a radial bearing of normal fit the impact varies in value from 0 through a maximum back to 0 during 180 deg., while in a thrust bearing the impacts are alike over the whole 360 deg. The relation between specific loads and bearing loads was also given for static conditions. But how can the endurance relation between the two bearings be estimated? If it be assumed that a certain number of impacts under any given load will use up a certain fraction of the available life of a bearing at this load, it can be shown by means of the fatigue formula that a thrust bearing under the same specific ball load as a radial bearing of normal fit, will only have one-sixth of the life (in number of shaft revolutions). If, however, a load is put on the balls in a radial bearing by expanding the inner race, the number of impacts will be equivalent to those in a thrust bearing for the same ball load. While, therefore, statically the load on a bearing increases with tightness for a given k, dynamically the increase in effective impacts will soon counterbalance the gain.

The slope of the fatigue curve is constant for any bearing, but its position will depend on many factors, such as material, type of bearing, conformity, and looseness, but it is of particular interest to note that for similar bearings the quality of the material is of paramount importance.

If the load tables for different bearings as given in the catalogs of different manufacturers be now considered, it is evident that for any load given a certain average life in total revolutions must be expected. The relation between loads given for different bearings may be more or less correct, depending on the material and the practical experience of the particular company, but the data can only have relative value because the service conditions are so different, even for the same estimated load. In some cases there is a variable load, in other cases an intermittent load, or possibly a

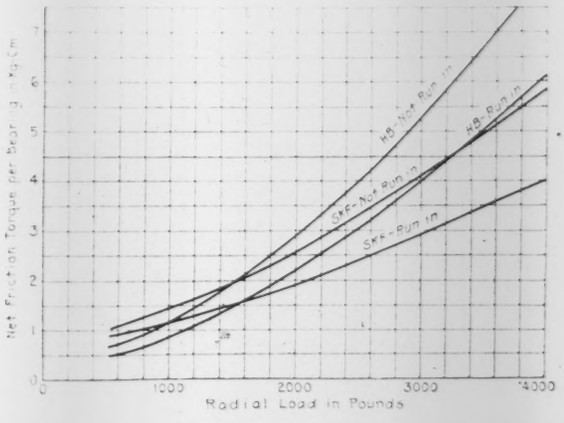


FIG. 6 RELATION BETWEEN FRICTION TORQUE AND LOAD IN BALL BEARINGS

(Medium-sized HB deep-groove and SKF spherical ball bearings, run in and not run in.)

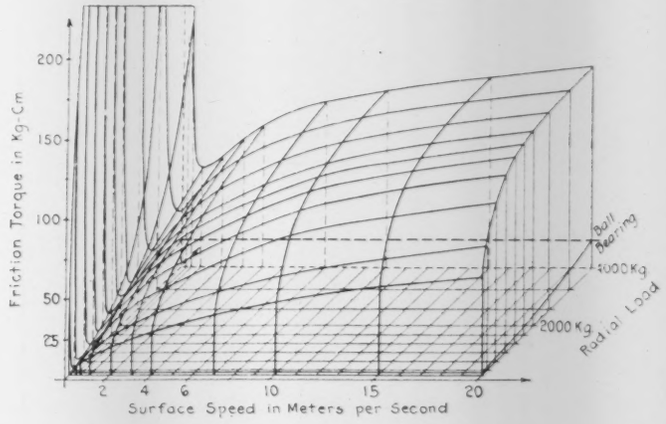


FIG. 7 COMPARISON OF FRICTION TORQUE IN PLAIN BEARINGS AND BALL BEARINGS FOR DIFFERENT SPEEDS AND LOADS

[Plain bearing 7 cm. in diameter by 23 cm. long, compared with ball bearing. Based on data from Stribeck and Lasche. Lubricant, heavy machine oil (absolute viscosity = 180 centipoises, approx.).]

continuous load. It is customary, therefore, to apply safety factors for the various cases.

In most catalogs the load ratings for different-sized bearings are based on Stribeck's formula $P = 0.2\ nkd^2$, although the values are corrected from experience. From the tests of the author's company

[1] Endurance-Test Data and Their Interpretation, by Heindlhofer and Sjövall, MECHANICAL ENGINEERING vol. 45, No. 10, 1923, p. 579.

it is found that the formula $P = 0.2\ nk_1d^{1.5}$ conforms better to the results of tests.

Under practical load conditions probably any bearing will fail from fatigue if given sufficient time; perhaps ten or a hundred years will be necessary. In the laboratory fatigue failure has occurred after more than four billion impacts. A characteristic fatigue failure is shown in Fig. 5. The starting point for such flaking is apparently just below the surface, for instance, 1/16 in. to 1/8 in. deep, depending on the load. Bearings in service can fail from many other causes, and particularly frequent are failures due to abrasion, which can happen in the surprisingly short time—less than an hour. When the lubricant is free from abrasive, no wear can be found in the bearing. Failures by indentation from foreign particles or by corrosion are also frequent. Careful handling of bearings is evidently necessary.

Having presented in the preceding paragraphs the load-carrying characteristics of ball bearings, it seems proper to also discuss briefly the most popular argument for the use of ball bearings, namely, their smaller friction loss as compared with plain bearings in general.

During the last two years a great number of tests have been made at the SKF Laboratory on the friction of ball bearings, and the coefficient of friction referred to the shaft diameter has been determined in relation to loads up to 4600 lb. and speeds up to 2000 r.p.m. Little variation has been found with the speed, but considerable variation with the load, although it so happens that when ball bearings are operated around catalog loads for medium speeds, the coefficient of friction referred to the shaft is nearly constant and near the minimum.

For other loads, particularly the lighter ones, the coefficient of friction gets higher and a choice is rather arbitrary and misleading, and a comparison between different bearings difficult.

Fortunately there is another and much simpler method of showing the friction loss in ball bearings, namely, by representing the torque, as directly measured, as a function of load and speed instead of calculating the coefficient of friction from the torque.

The torque varies little with the r.p.m., but is very nearly proportional to the load. (See Fig. 6.) What is more interesting is that for a given external load the torque is nearly independent of the size of the bearing, at least for medium sizes. For a deep-groove bearing the torque increases with the load somewhat faster than it does for a spherical ball bearing, and the torque of the spherical roller bearing is only about 50 per cent higher than for the ball bearing.

The lubricant influences the friction, but soap greases and light or medium oils give about the same torque, while an oil with greater viscosity at the operating temperature increases the friction considerably.

In order to obtain a comparison with the friction of plain bearings, the torques determined for various loads and speeds by Stribeck and Lasche have been plotted and compared with the torque for a ball bearing under the same load. (See Fig. 7.)

It is seen that for low speeds of the plain bearing, provided of course there is still fluid lubrication, the friction is not much greater than for a ball bearing. For extremely high loads and high speeds, when the torque for fluid lubrication also becomes more constant, the two types of bearings again approach each other in friction. But for intermediate conditions the friction loss in ball bearings will be materially less than in a plain bearing. One point in particular is worth while emphasizing: namely, that the starting torque of a plain bearing is usually from 20 to 40 times higher than the running torque, while it is only twice as high on a ball bearing.

The author is greatly indebted to his associates in the laboratory, Dr. Heindlhofer and Messrs. Sjövall, Dipper, Titus, and Walp, for the care and interest shown in this work. They have made it possible to arrive at the results here published.

A Chart for Determining the Weight of Moist Air[1]

By JOHN E. YOUNGER,[2] BERKELEY, CAL.

The calculation of the weight of air, taking into account its temperature, pressure, and moisture content, is laborious and has received the attention of many engineers. In this paper the theoretical basis of such calculations is clearly set forth, and a chart is presented which is perhaps the first simple and convenient graphical method to be devised for the calculation of such problems. It should be of great value to all those having to make calculations on the performance of fans, in heating and ventilating problems, and in aerodynamic experimentation in the wind tunnel.

THE reduction of quantitative data in heating and ventilating experiments and in wind-tunnel investigations is attended with a large amount of labor. The calculation of the weight of air is no small item in such work. It has been suggested that much time and labor might be saved if the weight of air could be determined from a chart at the time of the experiment. The accompanying chart is the result of considerable study of this question. It takes into account the temperature of the air, its hygrometrical conditions, and the barometric pressure, and gives the weight in pounds per cubic foot.

It is the purpose of this paper to review the theory and derive the formula on which the chart is based, and to explain its construction

Theoretical Basis of Calculations

The weight of a cubic foot of moist air is the sum of the weights of the dry air and the water vapor present. According to Dalton's law, the total pressure of a mixture of gases is the summation of the partial pressures of the constituents. The dry air and the water vapor may be assumed to follow the perfect-gas law. Expressing these statements in the form of equations, we have:

$$W = W_a + W_v \quad [1] \qquad P_b = P_a + P_v \quad [2]$$

$$W_a = \frac{P_a}{R_a T} \quad [3] \qquad W_v = \frac{P_v}{R_v T} \quad [4]$$

in which W = weight of 1 cu. ft. of moist air, lb.
W_a = weight of dry air in 1 cu. ft. of moist air, lb.
W_v = weight of water vapor in 1 cu. ft. of moist air, lb.
P_b = barometric pressure, in. Hg
P_a = partial pressure of dry air, in. Hg
P_v = partial pressure of water vapor, in. Hg
R_a = gas constant for air = 53.34
R_v = gas constant for water vapor = 85.8
T = atmospheric temperature, deg. fahr.

Substituting Equations [3] and [4] in Equation [1], and eliminating P_a with Equation [2], gives

$$W = \frac{P_b}{R_a T} - \frac{P_v}{R_a T}\left(1 - \frac{R_a}{R_v}\right) \quad [5]$$

It may be noted that the first term of the function is the weight of dry air at barometric pressure, and the last term is the weight subtracted when moisture is present, the sign of the term being determined by whether the ratio of the gas constants is less or greater than unity. Moist air is therefore lighter than dry air.

In Equation [5] the gas constants are known, and the temperature and barometric pressure may be determined directly with instruments. The determination of the vapor pressure requires further consideration.

An inspection of Equation [4] shows that the vapor pressure is proportional to the weight of the vapor present in 1 cu. ft. However, the temperature determines the amount of vapor the air will absorb. When the maximum amount of moisture is present for any temperature, the air is said to be "saturated" with moisture. Under this condition the water vapor exerts its maximum pressure for the fixed temperature; the vapor, which is really steam, is there-

[1] Thesis presented in competition for the 1923 A.S.M.E. Student Prize.
[2] Instr., Dept. of Mech. and Elec. Engrg., Univ. of Cal.

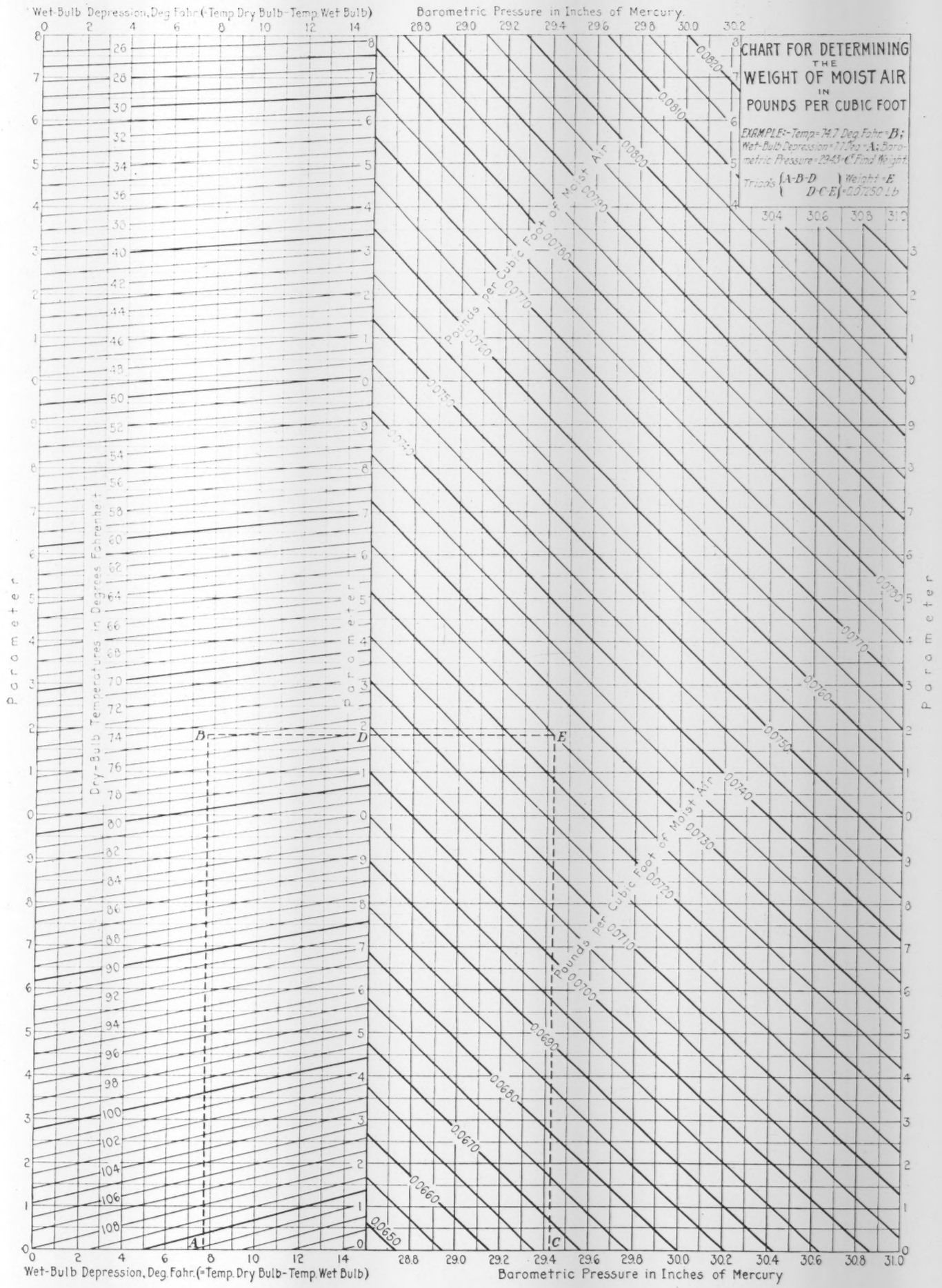


CHART FOR DETERMINING THE WEIGHT OF MOIST AIR IN POUNDS PER CUBIC FOOT
EXAMPLE:- Temp = 74.7 Deg. Fahr. = B; Wet-Bulb Depression = 7.7 Deg. = A; Barometric Pressure = 29.43 = C. Find Weight.
Triads {A-B-D, D-C-E} Weight = E = 0.07250 Lb
Wet-Bulb Depression, Deg. Fahr. (=Temp. Dry Bulb-Temp. Wet Bulb)
Barometric Pressure in Inches of Mercury
Dry-Bulb Temperatures in Degrees Fahrenheit
Pounds per Cubic Foot of Moist Air
Parameter
A
B
C
D
E

fore saturated, and the vapor pressure may be found from tables of the properties of saturated steam.

Air seldom contains this maximum amount of moisture. The ratio of the actual amount of moisture present to the amount that would be present at saturation is expressed as the "relative humidity." When the relative humidity is below unity, the air has a drying effect; that is, it will absorb moisture from whatever source present. Considering the process from the standpoint of the water source of the moisture, we say that the water will evaporate. This change in state from a liquid to a vapor requires a certain amount of heat, which is taken from the liquid or immediately surrounding material, thereby cooling them. The cooling property is made use of in the psychrometer, the psychrometric conditions of the air being indicated by the comparison of two similar thermometers, one with wet muslin wrapped around the bulb, and the other with the bulb open to the air. The relative humidity, and hence the vapor pressure, may be expressed as a function of the wet-bulb depression. The theoretical formula deduced by both Maxwell and Stefan upon the principles of the interdiffusion of gases, is:

$$P_v = P_t - AP_b(T - t) \quad \ldots\ldots [6]$$

in which A = an experimental constant, t = wet-bulb temperature, and P_t = vapor pressure at saturation for the temperature of the wet bulb.

Professor Ferrel further considered the equation for the U. S. Signal Service and conducted numerous experiments to evaluate the constant. The formula given was

$$P_v = P_t - 0.000367\ P_t\ (T - t)\left(1 + \frac{t - 32}{1571}\right) \quad \ldots\ldots [7]$$

the units being in inches of mercury and degrees fahrenheit.

This formula was used in computing extensive psychrometric tables published by the U. S. Department of Agriculture, and also in computing the Smithsonian Hygrometrical Tables published in the Smithsonian Miscellaneous Collections, 1918–1919.

Substituting this value for the vapor pressure in Equation [3], and expressing all pressures in inches of mercury and all temperatures in degrees fahrenheit, the weight per cubic feet of moist air under all conditions of humidity is:

$$W = \frac{70.73}{R_a(460 + T)}\left[P_b - \left\{P_t - 0.000367\ P_b\ (T - t)\left(1 + \frac{t - 32}{1571}\right)\right\}\left(1 - \frac{R_a}{R_v}\right)\right] \quad \ldots\ldots [8]$$

This equation contains three independent variables, P_b, T, and t; and two dependent variables, W and P_t. P_t is a function of t only, so that a chart of the equation would represent W as a function of the three independent variables.

Construction and Use of Chart

The idea of charting the psychrometric conditions of the air is not new. A number of such charts may be found in engineers' handbooks and in technical magazines, especially those devoted to heating and ventilating. However, they are designed to give data such as relative humidity, weight of moisture per cu. ft., total heat, etc. and not to give the weight of the air. The general method used in constructing these charts is to plot the weight of moisture per cu. ft. of air as the ordinate against dry-bulb temperatures, using rectangular coördinates. The series of weights and temperatures corresponding to fixed values of wet-bulb temperatures are plotted, giving lines of wet-bulb temperatures. In a similar manner, lines of constant relative humidity are plotted. Only one value of barometric pressure is used. An instrument, commonly known as the "hygrodeik," which is designed to give relative humidity by a graphical chart, is also constructed in this manner, the dry- and wet-bulb thermometers taking the place of the corresponding temperature scales.

In the accompanying chart another variable is introduced—the barometric pressure. This obviously necessitates the introduction of a parameter. The parameter to use, the arrangement of the variables, the scales, and the type of coördinates, logarithmic or rectangular, were arrived at by a study of the relative influence of the three variables on the weight of air, and the endeavor to acquire maximum simplicity in the details of drafting. The problem was largely one of trial and error.

The psychrometric division at the left of the chart was first constructed for a barometric pressure of 30 in. of mercury. To obtain the lines of constant dry-bulb temperatures it was of course only necessary to hold the dry-bulb temperatures constant and plot the weight of air against wet-bulb depression. The interval between the dry-bulb-temperature lines along any line of depression is almost constant. In fact, a table of air weights carried out to the fifth decimal place does not reveal any constant variation in the increments of weights between regular intervals of temperature. Likewise the curvature of the lines of constant dry-bulb temperatures is not perceptible in tables carried only to the fifth decimal place. The accuracy gained in the ease and uniformity of drafting more than compensates for the accuracy sacrificed in considering the intervals and slopes constant. This is, however, no fault of the design. Evidently, if the chart is made large enough and tables are carried out with sufficient accuracy, any desired degree of accuracy in curvature and spacing may be attained by plotting.

In building the barometrical division of the chart, one assumption is made: that is, that the increase in weight for the increase in wet-bulb depression, the slope of the dry-bulb-temperature lines, does not change with the barometric pressure. This is equivalent to assuming that the increment of weight added to saturated air for any particular wet-bulb depression is the same for any fixed value of dry-bulb temperature, regardless of the barometric pressure. If the error in this assumption is negligible, as will be shown later, the parameter, instead of giving weights for only a barometric pressure of 30 in. Hg. may be assigned other values of weights to correspond to other barometric pressures. These values, of course, must be assigned along the ordinate of the particular pressure.

The error in this assumption may be shown as follows: Let w be the weight added to the weight of saturated air for any particular value of wet-bulb depression. By subtracting the weight of saturated air from the weight of unsaturated air, we get the following expression for w:

$$w = \frac{70.73\left(1 - \frac{R_a}{R_v}\right)}{R_a(460 + T)}\left[P_b - P_t + 0.000367\ P_b\ (T - t)\left(1 + \frac{t - 32}{1571}\right)\right] \quad \ldots\ldots [9]$$

By taking the partial derivative of w with respect to P_b,

$$\frac{\partial w}{\partial P_b} = 0.000367\ (T - t)\left(1 + \frac{t - 32}{1571}\right)\left[\frac{70.73\left(1 - \frac{R_a}{R_v}\right)}{R_a(460 + T)}\right] \quad \ldots\ldots [10]$$

As the useful range of the chart is between the barometric pressures of 29 and 31 in. of mercury, the maximum change from the barometric pressure of 30 is 1. The partial derivative then expresses the error. For an average maximum error, assume T = 100 deg. fahr. and $(T-t)$ = 15 deg. fahr. Substituting these values, together with the values of the gas constants, in Equation [10], the error is found to be 0.000005 lb., or well in the sixth decimal place. This error is beyond the range of most hygrometrical tables, and may be completely ignored in charting. Values may be assigned to the parameters, therefore, so that the hygrometrical division of the chart may fit the various barometric pressures. The method of procedure was to assume basic weights, such as 0.076, 0.077, etc. and plot the lines for values of dry-bulb temperatures, wet-bulb depression, and barometric pressure. The first two mentioned were of course combined and the result carried by the parameter. With these major lines plotted and tested for accuracy, interpolation was made along lines of constant barometric pressure, with dividers for lesser divisions.

The size of the original chart was 28 in. by 20 in. The tables used chiefly were those by the U. S. Bureau of Construction and Repair for testing the ventilation of ships. With a chart of the size used, the accuracy can easily be brought within 0.1 per cent.

If a large chart of this nature be mounted on a drawing board and a T-square and triangle be used to follow up the coördinates, the speed in its use may be largely increased.

Technical Features of High-Speed Newsprint Manufacture

By C. W. MORDEN,[1] WEST LINN, ORE.

ONE of the basic industries of the Northwest Pacific States which is closely related to their forests and to the other forest industries, logging and lumbering, is the manufacture of paper. In this territory are some of the most modern paper-making plants in the country, and one of the companies operating in it, which also has mills in California and British Columbia, is rated as the second largest producer of paper in the world.

It is not, however, the intention of the author to deal with the paper industry of this region as a whole, but principally with an element in it, and a most important one—the high-speed newsprint-making machine. The present-day machine of this type has followed the general lines of the machines which preceded it, but in many of its features the developments of the last four or five years have been particularly pronounced.

Capacity of Large Newsprint Machines

To those who may not be familiar with the size and production capacity of a modern newsprint machine, it may be of interest here to state that what is termed a 164-in. high-speed newsprint machine occupies a floor space 270 ft. in length by 17 ft. in width, this width being exclusive of the machine's drive. The manufacturing process employed in such a machine, briefly stated, consists first of the continuous formation of the pulp fibers into a sheet of paper, which formation takes place in the Fourdrinier part of the machine, on what is known as the "wire." Following its formation on the wire, the sheet is subjected to a series of continuous pressing operations between press rolls, for the purpose of removing water and compacting the sheet. Following these pressing operations, the sheet is passed around the surface of a series of revolving drier cylinders, from which it passes to a stack of highly polished calender rolls, which give the sheet its finish. After leaving the calender, the sheet is wound into a wide roll on what is known as the "reel." This wide roll is subsequently rewound on what is known as a "winder." In this rewinding operation the sheet is slit by rotary slitters into widths corresponding to the length of the finished rolls of paper. It is from the winder part of the machine that the finished rolls are delivered.

At a speed of 1000 ft. per min. such a machine will turn out in a 24-hour day a sheet of paper 12 ft. 8 in. in width and approximately 250 miles in length, which expressed in tonnage amounts to between 85 and 90 tons.

The Stock and Its Preparation

The wood fiber, or pulp, which is converted into paper on this type of machine is of two kinds: ground-wood pulp, which is also known as "mechanical pulp," and sulphite pulp, which is sometimes known as "chemical pulp." The ground-wood pulp is produced by wet-grinding the pulp wood into fiber, and the chemical pulp is produced by reducing the pulp wood to chips which are made into a pulp by a chemical process in which the chips are cooked in a bisulphite cooking liquor under pressure. The fibers so produced are screened in special-type pulp screens, separate screens being used for each class of fibers. Throughout the screening operations, and in fact up to and including the time of the introduction of these fibers into the paper machine, they are suspended in water, which is made to serve as a conveying medium for them. The usual proportion of fiber to water is 3½ to 4 lb. of dry fiber per 100 lb. of liquid. When the fiber is being screened, or when it is ready to be fed to the wire of the paper machine, the quantity of water is increased to a point where the amount of fiber in proportion to the liquid is approximately ½ lb. of fiber per 100 lb. of liquid.

Previous to being fed to the wire of the paper machine, the ground-wood and sulphite fibers are mixed in proportions of approximately 25 per cent sulphite and 75 per cent ground wood, color is added and the mixed fibers, or "stock" as the papermaker calls them, are pumped to the paper-machine stock-storage chest.

From this chest the stock is pumped to the "head box" of the paper machine. This box is so arranged that a part of the stock pumped into it may be passed through a slide valve to the paper machine. This slide valve is known as the "stock gate" and is the control valve by means of which the amount of stock delivered to the paper machine is regulated. Due to the fact that the head above this valve is maintained constant, the quantity of stock passing through the valve also remains constant for any given gate opening, provided its consistency does not vary. In other words, there is encountered here an application of the hydraulic formula, $Q = AV$, where V, the velocity of flow, is kept constant.

Flow of Stock through Machine

The 3½ per cent consistency stock passing through this stock gate and drops through a pipe to the suction side of a centrifugal pump known as a "fan pump," where it mixes with enough water to reduce its consistency to approximately 0.6 per cent. This mixture of stock and water is pumped to a battery of screens just ahead of the paper machine. From these screens the stock flows by gravity to the "flow box" of the paper machine. This flow box is arranged with baffles the function of which is to level out the flow of the 2300 or 2400 gal. per min. of stock mixture as it flows on its way to the paper-machine wire.

The outlet end of this flow box is closed by a brass plate known as a "slice." This slice serves as a dam to hold back the stock mixture in the flow box, and also as a valve to control the flow of stock to the paper-machine wire. This slice is arranged so that it may be raised or lowered with reference to the bottom of the flow box, which at the point where the slice is located is known as the "apron board." An orifice is thus provided between the bottom of the slice and this apron board, through which the stock flows to the wire of the paper machine. A new and patented development in connection with this slice is the provision of a vertically adjustable flexible lip along the lower edge of the slice, by means of which the cross-section of the opening between the bottom of this lip and the upper surface of the apron board below it may be varied at any point across the opening. This makes it possible to control the flow of stock at any point below the slice and thus to control the distribution of this stock to the paper-machine wire. This type of slice is practically the equivalent of a gate valve with a gate 13 ft. wide and so arranged that its lower edge is vertically adjustable at any point along its length.

By raising or lowering the slice the velocity of flow of the stock to the wire can be controlled. This is due to the fact that such raising or lowering changes the orifice area under the slice, and as this area is decreased or increased the head of stock behind the slice raises or lowers, which change in head brings about a change in the spouting velocity of the stock passing through the orifice in accordance with the hydraulic formula $V^2 = 2\,gH$.

As this stock spouts out from under the slice it is delivered on to the upper surface of a fine-mesh woven-wire belt which is traveling in the same direction as the stock. This wire belt, or "wire," is 164 in. wide in the case of the particular machine under consideration.

Formation of the Sheet

Spouting the stock on to the wire in this way is a characteristic of high-speed-machine operation. It has been the object of much study and experimentation for it plays a very important part in the formation of the sheet of paper on the paper-machine wire. By "formation" is meant the interlacing and uniform distribution of the pulp fibers on the surface of the paper-machine wire.

It is essential to the proper formation of the fibers that they shall be suspended at the time of formation in a large volume of water.

[1] Mill Manager, West Linn Plant, Crown Willamette Paper Co. Assoc. Mem. A.S.M.E.

For presentation at the Portland Regional Meeting of The American Society of Mechanical Engineers, Portland, Ore., June 22–25, 1925.

In the case of a 164-in. machine producing paper at the rate of 1000 ft. per min., a total of 19,600 lb. of water per minute is delivered to the wire, and in this quantity of water is carried the 118 lb. of pulp fibers which are delivered to the machine every minute. Most of this water, namely, between 18,800 and 18,900 lb. per min., passes through the wire, and in doing so leaves the wet pulp fibers on its surface. Various agencies assist in the removal of this water, such as the capillary attraction of the table rolls over which the upper surface of the wire runs, suction boxes below the wire, and the action of the "couch" rolls at the end of the wire. Space does not permit discussion of these details.

The water which passes through the wire collects in a pit underneath it, whence it flows to the suction side of the fan pump previously mentioned. It is at the inlet to this pump that the pulp fibers going to the machine are mixed with and diluted by the water from the wire pit. It is the fan pump's function to deliver this mixture of water and fibers to the machine, and thus it is seen that it is this water which by its circulation serves to deliver the pulp fibers to the wire and which serves as a medium to suspend them while they are in the process of being formed into a sheet of paper on the wire.

This formation takes place very rapidly, for from the time that the stock first touches the wire to the time it leaves it in the form of a sheet of paper, only approximately $2^1/_2$ seconds elapse.

Removing Water from the Sheet

The sheet thus formed by the wire passes on to a woolen belt called a "felt," by which it is conveyed between a pair of rolls which press out the water. From the felt of this first press the sheet is passed to the felt of a second press, and thence to that of a third press. The sheet of paper as it enters the press section of the machine contains from 86 to 88 per cent of water, and as it leaves, from 68 to 72 per cent. The percentage of water in the sheet as it leaves the press section is very important, as it determines the amount of water which will have to be evaporated from the sheet by steam in the drier rolls. It costs roughly fourteen times as much to remove water from the sheet in the driers as it does by means of the mechanical methods employed at the presses, so it is natural that more and more attention is being paid to studying mechanical methods of water removal. On the press section of the machine this is being done by installing dynamometers to measure the pressure applied by the presses, and similar dynamometers are installed to measure the tension applied to felts. The actual work done by the presses in removing water is checked by measuring the water removal at each press. Supplementing this the manufacturers of the woolen felts are studying the construction and material of their felts very carefully; in fact, one of the manufacturers of these felts has developed and patented the methods above referred to for measuring the work done by the presses.

In passing the sheet of paper from the wire to the first felt and from the first felt to the second, etc., use is made of compressed-air jets of special construction which blow the sheet across the short distance from one felt to the next. Another method is to bring a small suction wheel into contact with the sheet on the wire, the action being such that the sheet is sucked from the wire and blown over to the first felt. These methods are patented and their development, as well as and that of other mechanical methods of handling the sheet between the various sections of the machine, is really what has made possible machine speeds of 1000 ft. per min. The old methods of passing the sheet depended entirely upon the skill and quickness of the machine operator and reached a limit at a speed well below 1000 ft. Another item of interest in connection with the passage of the sheet through the machine is that the sheet is stretched approximately 10 per cent, due to the pull exerted on it as it passes from one section of the machine to the next.

Individual Motor Drive for Each Section of Machine

Another development which has made possible speeds of 1000 ft. per min. and over is the development of an individual electric-motor drive for each section of the machine. This has eliminated the old mechanical drive consisting of lineshaft in the basement with belt drive on cone pulleys to mortise-gear units on the machine floor. The special feature of the individual motor drive centers about the means for controlling the speed of all motors so that a constant speed is maintained by each section of the machine. Any variation of speed would tend to break the sheet of paper as it passes from one section to the next. The principle used here in the case of the particular machine being discussed is to provide direct-current motors for driving each section of the machine, to each of which is belted, through cone pulleys, an alternating-current synchronous motor of one-fifth the horsepower of the main driving motor. These synchronous motors are all connected to a common bus and all therefore run at exactly the same speed, which is a characteristic of synchronous equipment of this kind when the units are operating on a common circuit. A pair of spur gears connects the cone-pulley drive of each of these synchronous motors to the d.c. motor with which it operates. This gear ratio is made different for each of which of the drives, and thus the d.c. motors are made to operate at a speed which corresponds to the required speed of the sheet of paper in each section of the machine. The slight varations in this speed which are necessary in order to provide the proper draw of the sheet between sections are made by varying the position of the belt on the cone pulleys which form the connecting drive between the d.c. motor and its a.c. motor. The constant-speed a.c. motor controls any tendency of the d.c. motor to vary in speed by adding load to the d.c. motor when it tends to speed up, and by supplying power to the d.c. motor shaft when the d.c. motor tends to slow down due to variations in its load.

The 250-volt direct current used by the nine motors on the machine is supplied by a turbine-driven direct-current generator. The exhaust steam from the turbine driving this generator is the steam used in the drier rolls to dry the paper. As the amount of steam required for drying is considerably in excess of that required to develop the power necessary to operate the machine, this necessitates either using live steam to make up the deficiency or providing a means of securing additional exhaust, or low-pressure steam. In the case of the machine under discussion, additional low-pressure steam is secured from the turbine by adding to its driven load an a.c. generator, the power output of which is delivered to the 550-volt a.c. buses which supply the a.c. motors throughout the mill. This arrangement makes it possible to vary the amount of steam to the turbine in accordance with the demand for steam by the paper-machine driers. The resulting variations in power from the turbine are taken care of by the a.c. generator, which delivers this surplus energy in the form of electric power to the a.c. distribution system of the mill. There is another advantage in this arrangement, which is that the a.c. generator serves as a speed governor for the turbine, supplementing the regular turbine governor.

In the drier section of the machine a mechanical method of threading the paper from one drier roll to the next has been developed. This replaces the old hand method and the result is accomplished by means of a rope carrier consisting of two cotton ropes running side by side on sheaves located at one end of each of the driers. The sheet of paper as it starts over the drier section is thrown in between these two ropes and is held between them and thus threaded through the full system of drier rolls.

As the paper leaves the driers, air jets are used in passing it to the calender stack. The same means are employed in passing the paper from the calender stack to the reel. The reel itself has been improved so that practically no paper is lost in changing the sheet from a completed reel to the mandrel on which the next roll of paper is being started.

In the winder section of the machine no marked changes have been made compared to the practice formerly used in the lower-speed machines. As stated earlier, the winding operation is that which completes the paper-making processes of the paper machine.

A Correction

ON PAGE 442 of Section Two of the May, 1925, issue of MECHANICAL ENGINEERING in the synopsis of Mr. Dow's paper on Mechanical Features Affecting the Reliable and Economical Operation of Hydroelectric Plants, it was erroneously stated that this paper was translated by Mr. L. C. Marburg. Mr. Klopstock's paper on Recent Investigations in Turning and Planing and a New Form of Cutting Tool, a synopsis of which appears on the same page was the paper which Mr. Marburg translated.

Some Freight-Car-Maintenance Problems

By C. G. JUNEAU,[1] MILWAUKEE, WIS.

UNDOUBTEDLY the greatest problem today in connection with freight-car maintenance revolves on the constant endeavor of the railroads to obtain the maximum use of the car unit. The utilization of cars is the product of their loading and their movement. Successful attempts are continually being made to load each car with greater tonnage, and to increase the distance each car is hauled per day. Again, wherever there is an opportunity to fit the trains to the traffic, this is being done so that the tonnage will be handled in fewer trains, and thus effect a saving in those transportation expenses that fluctuate with train mileage. The trend of this endeavor on the part of the railroads to obtain greater use of their cars and trains is shown in Fig. 1.

Along with this steady increase in the utilization of the freight car has gone a steady change in the construction and strength of

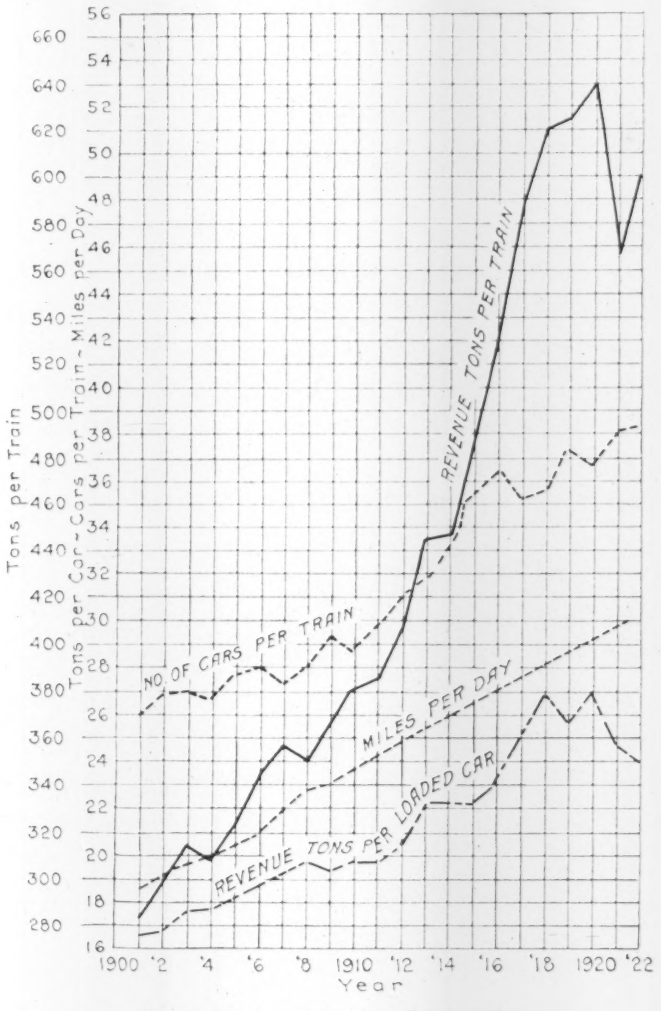

Tons per Train
Tons per Car - Cars per Train - Miles per Day
REVENUE TONS PER TRAIN
NO. OF CARS PER TRAIN
MILES PER DAY
REVENUE TONS PER LOADED CAR
1900 '2 '4 '6 '8 1910 '12 '14 '16 '18 1920 '22
Year

FIG. 1 UTILIZATION OF THE FREIGHT CAR

the car. The all-wooden car is fast disappearing and its place is being filled by the composite car and the steel car. During the past twenty years the average carrying capacity of a freight car has increased from 29.4 tons in 1903 to 43.1 tons in 1922. The light weight of cars has also increased. The all-wooden box cars acquired in the 1890's weighed from 14½ to 16 tons, whereas the composite box cars acquired in 1919 weighed 23½ tons. Fig. 2 shows the replacement of the all-wooden freight car by the composite and the steel car. Fig. 3 shows the increase in the average carrying capacity of freight cars of the country and pictures the trend of increased light weight of cars as shown in the case of box cars.

These changes in the weight and capacity of the freight car have resulted in its increased punishment. Although freight cars have been greatly strengthened to meet the additional burdens placed upon them, and while there is really no disagreement throughout the country covering the strength requirement of these cars, the railroads do, however, continue to have bad-order cars in great numbers. What is actually occurring is increased punishment to freight equipment because of the lack of cushion formerly provided in train movement by reason of the wooden car's disappearing. If operating methods are to proceed as in the past, there is absolutely no question but what resilience will have to be furnished through the medium of a suitably designed draft rigging, or else we may expect distinct failure of parts following a progressive cycle, depending upon considerations of relative strength factors.

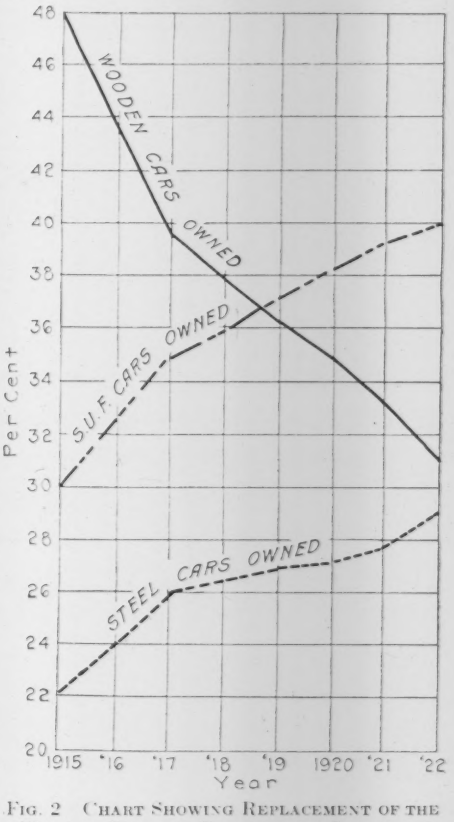


FIG. 2 CHART SHOWING REPLACEMENT OF THE WOODEN FREIGHT CAR BY THE COMPOSITE (STEEL UNDERFRAME) AND THE STEEL CAR

Furthermore, while the freight car as a whole has been designed and constructed to meet modern operating conditions, many of its parts are not equal to the tasks imposed. The draft gear seems to be about the weakest part of most cars, when, as aforementioned, it must be depended upon to lessen the punishment received by the car. Very little has been done to improve the foundation brake rigging and air-brake apparatus on the modern car as compared to the old unit, and yet experience shows that air brakes are responsible for approximately one-fourth of the transportation delay to cars in bad order in train yards and on running-repair tracks. Railroads cannot well continue accepting this kind of a performance if they desire to keep the modern car in continuous service a maximum amount of time.

IMPORTANCE OF DESIGN AND CONSTRUCTION

The design and construction of freight cars was never of more importance than it is today. Each acquisition of new equipment should be made the subject of special study by the mechanical and transportation officials to determine, first, the actual necessity for the cars, second, the size and capacity which will give the greatest net returns to the company, and third, the type of car. A thorough study of the design should be made before any cars are built. It is undoubtedly true that maintenance expenses are considerably increased due to failure to take everything into consideration when preparing the designs. The only way to insure that new cars when acquired will be a credit to the mechanical department, is to keep designs of each type constantly under way. Each detail must be critically analyzed and compared to existing types, and

[1] Master Car Builder, C. M. & St. P. Ry. Mem. A.S.M.E.

Contributed by the Railroad Division for Presentation at the Spring Meeting, Milwaukee, Wis., May 18 to 21, 1925, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. All papers are subject to revision.

an endeavor made to eliminate its defects. A comparison of the final result with the original will usually show a surprising number of changes, and will convince any one of the necessity of giving long and painstaking attention to every new design.

Few cars designed today can be criticized as lacking in strength in the essential parts, but it is also important that cars should be light so they can consistently be made easy to repair, and well protected against corrosion. Few cars meet all these requirements, and their failure to do so has led to severe criticism of car design from a maintenance and operating standpoint.

In the matter of designing and constructing a modern car, the details of material, as previously mentioned, cannot be judged entirely from strength requirements. The question of deterioration over a period of years needs to be given serious thought, because, for instance, if a roof becomes defective in one-half the time that the remainder of the principal portions of the car run before needing attention, and has to be repaired, the car as a whole is held. This emphasizes the need of studying the parts in relation to their cycles of renewals, so that they may be grouped and the number of days they are detained on the repair tracks consequently reduced.

Sources of Deterioration of Cars

One of the greatest sources of deterioration of freight cars, irrespective of the material or construction, is corrosion or decay, which continuously exerts its destructive influence whether the equipment is in service or not. In the case of wood parts there are very few renewals which are not directly caused or at least greatly hastened by decay. A close analysis of failures which appear to be purely mechanical will generally disclose the gradual destruction of the piece by decay as the original source of weakness. The matter, therefore, of preventing the destruction of material—by chemical action in the case of metal and by the propagation and growth of the destroying fungi in the case of wood—is worthy of considerable study.

A car must also be considered from the standpoint of items affected by friction or transportation such as wheels, brake shoes, draft rigging, couplers, and the trucks as a whole. The wear and failure of these parts constantly increase with the greater utilization obtained from the modern car. To withstand this greater hardship it is essential that proper designs and proper material be employed in each case. The use of correct metals and alloys requires a more thorough study. Certainly we never should use a brittle steel for parts subject to great shocks, or soft iron for parts subject to much friction, yet this has been done, resulting in failures. Sills, framing members, etc. are subject to strains that in time wear them out. These parts must be of such construction that shocks and vibrations incident to the service will not impair their efficiency.

Consideration should be given to building a car strong enough to meet general requirements and yet so put together, materials and otherwise, as not to be unduly heavy. In other words, cars should be designed to have a high ratio of load to total weight. This is important because dead weight is a factor in train movement. A five-year average of the freight trains handled on one western road showed that for each net ton of revenue freight carried the train hauled 1.2 tons of dead weight, exclusive of locomotive and tender. Approximately 66 per cent of the cars hauled were loaded. It is a well-known fact that automobile manufacturers have made successful attempts to reduce the weight of automobiles, yet retaining or increasing their loading capacity. Certainly this matter needs very serious study on the part of car designers.

It is not a good policy to materially sacrifice the design of a car to accommodate details. Standards, however, are unquestionably excellent. At the present time details that will absolutely interchange between all different cars are too few. As long as cars are built for some particular service we may expect to have a difference in construction due to the commodity to be transported. The present tendency is toward refinement in car design, and if it continues, equipment built a few years hence will probably be radically different from what is considered standard practice today. Already a remarkable number and variety of new designs of cars have been introduced.

In the matter of maintaining freight cars it must be continually kept in mind that cars that can be kept in continuous service with a minimum cost of maintenance and which are sufficiently efficient to protect the lading in transit, mean dollars and cents to the railways. One road found that during a five-year period the cars on its line averaged between 25 to 56 days per year in bad order; also that the frequency of repairs averaged between 22 to 34 times per year. In other words, a defect occurred for about every 500 miles the car moved.

Classes of Freight-Car Repairs

Freight cars are subject to many defects which make them unfit to operate. Some are of little consequence; others require considerable time and labor to repair. In general, freight-car repairs are governed by frequent renewal of certain parts and infrequent renewal of other parts. They are roughly divided into two classes: lights and heavies. Light repairs consist of work done to offset current wear, breakages, and loss of parts accruing from ordinary handling and movements of cars day by day. Heavy repairs

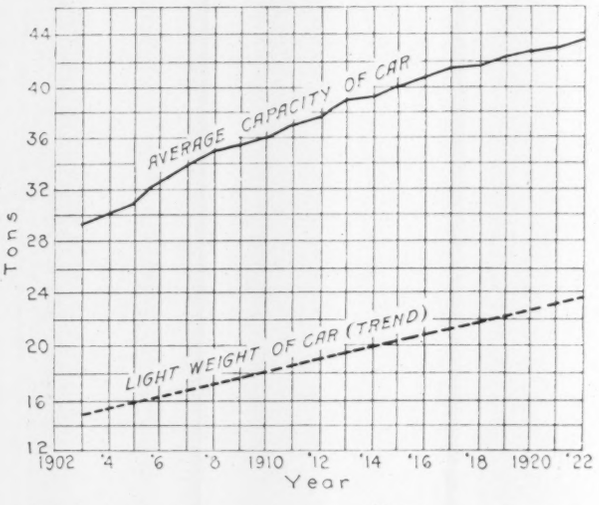


Fig. 3 Chart Showing Increase in Capacity and Light Weight of Freight Cars

accrue, generally speaking, from four different sources: wrecks, ordinary wear and tear as accumulated over a period of years, natural deterioration, and obsolescence.

To a certain extent the problem of handling light repairs is readily dealt with; that is, given reasonably adequate supplies of tools, material, and standard parts, and a certain force of men, that part of the maintenance problem will take care of itself, and cars will be switched on and off of the rip track daily without much fluctuation. It has generally been found that six light-repair cars are repaired during the day to every one left over at the end of the day, and it is this feature of the work which results mostly in an increase or decrease in the bad-order-car situation.

The situation with respect to heavy car repairs is entirely different. Repairs to cars due to wrecks and accumulated ordinary wear and tear are usually accomplished by replacements in kind, but more extensive work is required to overcome obsolescence as in this case it becomes necessary to strengthen and remodel the cars to overcome inherent weakness in design and construction. These cars remain a comparatively long time out of service and require a comparatively large expenditure to place them in proper condition. It is in the handling of heavy car repairs, therefore, that the greatest opportunities exist to produce economies and reduce the time that the cars are held out of service.

Frequency of Repairs—Average Age of Equipment

Studies made of maintenance policies and work done in repairs over long periods, particularly in relation to the life of equipment, indicate that under conditions prevailing for the last thirty years freight cars have been given heavy repairs by owners on an average of once every eight years. In other words, at the end of the eighth year the car, if normally maintained in the meantime so far

as wearing parts are concerned, would require heavy repairs, which would again make it good for another cycle of about eight years; after which it would be again given heavy repairs for another eight years of service if found to be of proper design, or would run with a limit of repairs until worn out. In this case the life would be extended to approximately twenty-four years or more, with a few years added to the life of the car where it was permitted to run in some minor service until entirely worn out. If at the end of the second cycle the car was found not worth repairing, it would be run about four years longer until worn out. In actual practice there is considerable variation of these figures, so that studies have developed that the average life of equipment has been approximately twenty years, particularly the body, there being a factor of safety required in the trucks which made them serviceable for about five years longer than the body. Practically all freight cars in the country have been depreciated on the basis of 20 years' life for the body and 25 years for the trucks, as set up by the Interstate Commerce Commission for valuation purposes.

There is no specific record of the average frequency of rebuilding or heavy repair work by classes of cars. The general data indicate an average period between heavy repairs to box cars of about nine years for those equipped with modern steel underframes, and about eight years for those with wooden underframes and relative construction. Modern refrigerator cars will require about seven years between heavy repairs because of the necessity of going over the insulating feature as well as overcoming structural deterioration. Stock cars will run about nine years due to their light construction. Coal cars will run from nine to ten years whether all-steel or composite. Steel floor sheets usually last about eight years and the side sheets about nine years, so that if false floors are used for a while the cars can be run about nine years. This has been the experience with the character of metal used in the past, which has been unable to withstand sulphuric acid coming from the coal.

The wisdom of having freight cars in good condition cannot be doubted, and the expense of maintenance is justified, particularly where the general condition of equipment is such as to require heavy work to overcome what we might call "inherent obsolescence." In other words, where the proper cycle of heavy repairs as previously explained has not been maintained in due course, there develops a degree of deferred maintenance, and to a certain extent, obsolescence, which must be overcome sooner or later.

In order to maintain a normal situation as to design, construction, and obsolescence, the average age of equipment should be carefully noted at all times, as this reflects whether or not old equipment is being retired as due and replaced with new or rebuilt equipment. It has been customary to assume an average life of twenty years for freight cars. Assuming that a road has an ownership of 100,000 cars normally at all times, then from year to year it should retire or rebuild an average of $1/20$ of 100,000 or 5000 cars and should acquire new equipment each year equal to the number actually retired and not replaced with rebuilt equipment.

It will be noted that a road has the choice of two policies in maintaining its equipment to overcome inherent weaknesses. It can retain its equipment for a longer period by overhauling and improving existing cars, or it can retire and replace the cars with new equipment at a more rapid rate. The policy to be followed depends upon many considerations, such as finance, operating expenses, and the extent of obsolescence.

The matter of appropriations for freight-car maintenance deserves special study. Operating revenues have many mouths to feed, and when heavy retrenchments in operating expenses become necessary it is to a certain extent inevitable that the maintenance departments bear more than their proportion of such retrenchments because maintenance work can be temporarily deferred without immediately destroying the effectiveness of the transportation machine. However, injudicious savings in maintenance result in actual loss to a road through increased transportation costs and heavier later costs to overcome deferred maintenance. The problem, therefore, is to determine where economy ends and loss starts in the retrenchment of maintenance expenses. An ideal condition would be to have the repair work equalized over the twelve months' period by arbitrary charges against income in the months of great business for credit and use during periods of revenue depression.

It is costly to make repairs to cars without proper repair facilities and equipment, and conditions become worse as the weight and capacity of cars increase. It is of course out of the question to provide numerous costly machines and tools where only a few cars are handled, but some repair tracks that turn out as many as 100 cars a day have practically no equipment for expediting the work. The labor cost of repairs made on large-repair tracks is usually much too high. Careful planning of major operations and the judicious expenditure for facilities will do much to correct this condition. This problem again is a financial one, and a road has the choice of three policies to pursue: first, resort to man power due to inadequate facilities; second, add a few machines or tools each month so as to gradually reduce the repair expenses; and third, spend a considerable sum to immediately equip the shops and repair tracks with the needed facilities.

Obviously the policy to be pursued will depend upon the road's finances, though no road can long afford to pursue the first policy for the lack of repair facilities decreases net earnings. Capital-account expenditures cannot be made to better advantage at this time in any direction by the railroads than through provisions for modern repair shops and equipment. In the past the amount of money expended by the railroads for logical repair-shop facilities has been lamentably out of proportion to the amount of money expended for cars and locomotives, resulting in inadequate facilities for needed repair and modernizing work. The conclusion is that every advance in the art of car design should be met with a corresponding advance in the facilities for maintenance.

There is every reason to believe that the traffic on railroads in this country will increase as rapidly in the future as it has in the past, and that the locomotives and cars will continue to grow larger. Bigger power is demanded at all times, and yet to haul a maximum train not only requires strengthening of freight cars, but roadways, bridges, roundhouses, shop facilities, and tools; and it must be remembered that when the freight cars are not used to the fullest degree they present a corresponding loss in all these elements, such as makes these extreme and uneven developments oftentimes a poor net result.

Every freight car involves not only the cost of maintenance but interest on the investment and current depreciation charges, both factors now being higher because of increased investment cost. The object of the design of cars, and the installation of facilities to make the maintenance cost relatively low, should be to increase the days of service by decreasing the days of detention in bad order because the greater present-day cost of maintenance, interest, and depreciation charges imposes upon proper railway management a greater availability for constant use.

Teaching Safety to New Employees

AN ILLUSTRATED safe-practice pamphlet entitled Teaching Safety to New Employees, has just been published by the National Safety Council. The pamphlet is non-technical in reading matter and is the combined experience of the industrial members of the Council. It is edited by 75 safety engineers who form a volunteer committee for such work.

"One of the most important considerations of industrial management is the introduction of the new employees into the plant," reads the introduction. Statistics show that the new employee is more liable to injury than one who has seen long service.

"Accidents to new employees are due not so much to carelessness or thoughtlessness as to lack of familiarity with the hazards and working conditions. Another factor is nervousness due to a desire to equal the production or speed of the more experienced workmen. Mechanical safeguards in specific industries may, to a certain extent, prevent accidents from both of those causes. The greater number of accidents however, occur from causes that are not preventable by guards. Education and supervision are the only effective methods of attack for such accidents.

"This makes it very important that special care be taken to teach the beginner. The experience of the National Safety Council shows that it is advisable to reach the new man just as soon after his employment as possible."

Inquiries regarding this pamphlet should be addressed to the National Safety Council, 168 North Michigan Avenue, Chicago.

SURVEY OF ENGINEERING PROGRESS

A Review of Attainment in Mechanical Engineering and Related Fields

AERONAUTICS (See also Internal-Combustion Engineering: Bristol Jupiter Aircraft Engine: Beardmore Crude-Oil Aircraft Motor; Marine Engineering: Kirsten-Boeing Propeller)

British Dirigible Design and Tests

THE building of ships of 5,000,000 cu. ft. capacity is under consideration in England. The *R-33* is being reconditioned at the suggestion of the Accident Investigation Sub-Committee of the Aeronautical Research Committee and of the Airship Stressing Panel in order to obtain further information on the subject. This ship was chosen because she is considered a sturdily built ship, being a copy of the German *L-33* and also because a series of wind-tunnel tests have already been made on a scale model of this airship. Thus a comparison of model and full-scale results should become possible, and should assist materially in indicating how far model figures can be applied to the full-scale airship.

Special apparatus has been devised for the purpose of the tests and the *R-33* has been fitted up with about 200 orifices, the pressure on which during various maneuvers will be measured and recorded simultaneously by electrical means on a number of special manometers. [It will be recalled as described in MECHANICAL ENGINEERING, vol. 45, no. 9, Sept., 1923, pp. 513–520, that the same method was used in the design of the *Shenandoah*.] Another series of tests that are planned for the *R-33* relate to ascertaining the mass of air which is moved with the airship, and for this test the ship has been equipped with pointers showing the helmsman the angle on his rudders, etc. The resulting motion of the airship in space will be obtained by photographing simultaneously with a moving-picture camera a number of instruments fitted in the control car, the reading thus obtained being further checked by flying the airship over a camera obscura on the ground, on the table of which the motion is recorded. (Editorial in *Flight*, vol. 17, no. 849/14, April 2, 1925, pp. 195–196, *g*)

The Pistolesi Variable-Pitch Airscrew

THIS airscrew is being developed in Italy by the company which built the *Savoia* aircraft. The propeller was recently fitted to a Spad-XIII engine and tests were made with the machine fitted with a variable-pitch airscrew and with a fixed-pitch airscrew. It is said that with a variable-pitch airscrew a gain was obtained of 5 m.p.h. on the horizontal speed, a gain of 2 min. or 15 per cent of the total time in climbing to 13,900 ft. elevation, and a gain of 1640 ft. on the practical ceiling. Tests made on the bench are said to have indicated the ability of the propeller to withstand mechanical stresses.

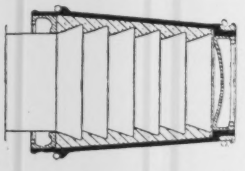


FIG. 1 JOINT EMPLOYED FOR SECURING THE WOODEN BLADES IN THE CENTRAL HOUSING OF THE PISTOLESI VARIABLE-PITCH AIRSCREW

Briefly the Pistolesi variable-pitch airscrew consists of a central metal portion carrying two wooden blades, the roots of which are set in steel muffs. The blades can be rotated within the central housing around their longitudinal axis by means of hand-operated gearing. The root of the blade (Fig. 1) has a series of conical steps formed on it which engage with similar steps formed inside the muff. The correct angle, number, etc. of these conical steps are factors which called for considerable experiment and research.

The Pistolesi airscrew is controlled by means of a small hand crank located within easy reach of the pilot, and only five or six turns are needed to obtain any pitch required for ordinary purposes. Complete reversal of the blades as regards pitch may be obtained with about 25 turns of the operating crank, and this may be accomplished by means of some form of friction gear from the engine. (*Flight*, vol. 17, no. 11/846, Mar. 12, 1925, pp. 145–146, 3 figs., *d*)

The Rohrbach All-Metal Flying Boat

DESCRIPTION of the type being developed by William Beardmore & Co., Ltd., British shipbuilders, at the company's Dalmuir shipyard.

In this boat duralumin is used throughout and no fabric is employed. The structure is built up of riveted girders and plates, which makes it possible to employ larger wing spans and higher powers than have hitherto been employed for flying boats of this class. The boat is fitted with collapsible masts and sails. It is of the monoplane type with cantilever wings arranged at an enhanced dihedral angle. A rectangular-shaped fuselage forms the hull of the ship and is equipped with metal stabilizing floats. The engines are located above the wings and raised clear of them by metal struts. The boat has a span over wing tips of 96 ft. 9 in. and weighs fully loaded 8140 lb., to which a useful load of 4400 lb. may be added. The boat is equipped with two-cycle Eagle-IX engines, each having twelve cylinders of $4\frac{1}{2}$ in. bore and $6\frac{1}{2}$ in. stroke. At a rated speed of 1800 r.p.m. the motors are designed to develop a total of about 720 b.hp. It is stated that maneuvering can be easily carried out with only one engine running. (*The Engineer*, vol. 139, no. 3613, Mar. 27, 1925, p. 350, 4 figs., and 2 figs. on p. 354, *d*)

Production of War-Aircraft Construction

THE question which arises is how England is to get high-speed pursuit planes in time of war. It is obvious that the country will have need of them in very large quantities, for the 56 squadrons or so which are promised six years hence for the Home Defense Force, and the odd 30 or 40 or 50 squadrons which might be formed at that time for the rest of the Royal Air Force (there are only 18 abroad at the present moment), would be neither here nor there in a great war in which the whole British Empire was concerned.

The next war is going to be a very much bigger affair than was the last one, and even then England had something like 200 squadrons, so it is fairly evident that instead of thinking about producing roughly a hundred squadrons six years hence, she ought to be laying the foundations for the production of something like 500 squadrons very possibly inside five years from now.

Now if one looks at all the various types of airplanes which are at present being produced for the Royal Air Force, leaving out all the old reconditioned wartime types and considering only those which are of recent design and are intended to re-equip existing squadrons, one finds that nearly all of them are constructed very largely of wood. And the one outstanding fact about mass production of aircraft in the event of a great war is that they cannot be built of wood for the all-sufficient reason that there is not enough wood obtainable in the world to build them.

The fact is that during 1917–1918 the U. S. Aircraft Production Board felled all timber which was suitable for aircraft and was within transportable range of rail or river. Some of it was used for making aircraft which never got as far as the War, and the rest of it has been used for frame houses or for firewood. And so there is none left.

Possibly if one could get at the timber in the middle of the great forests in America, including Canada, the wood exists. But there is no earthly way of getting it out and getting it cut up and shipped in sufficient quantities to European or even to American manufacturing centers in time for the initial stages of a great war.

First of all the timber would have to be discovered, then railways would have to be laid for hundreds of miles over trackless country into the forests themselves to get the timber away from where it

grows. That process together with the later process of handling would take years. And during those years the design and construction of all-metal or almost-all-metal aircraft would have progressed so rapidly under stress of war necessity that England would not want the timber when secured.

Consequently it seems that the perfectly obvious thing for the Air Ministry and the American Air Services to do is to start right now on the assumption that wooden aircraft are not going to be used in the next war. If once that idea is accepted as an axiom, then England can get along with the design and construction of metal aircraft. Part of a serial article entitled, On Imperial Safety, in *The Aeroplane*, vol. 28, no. 15, Apr. 15, 1925, pp. 349–350, *g*)

ENGINEERING MATERIALS

Pelumina Cement

DATA of a new cement developed by G. & T. Earle, Ltd., Wilmington, Hull, England, which is claimed to have a sufficiently slow initial set to enable the mortar to be thoroughly and consistently mixed and placed in position around the reinforcing parts; but once in position the concrete hardens quickly so that in three days its strength about equals what was previously expected from concrete in 28 days.

The new cement is of the portland type, but varies from the usual composition in that the alumina content is increased. Also a higher

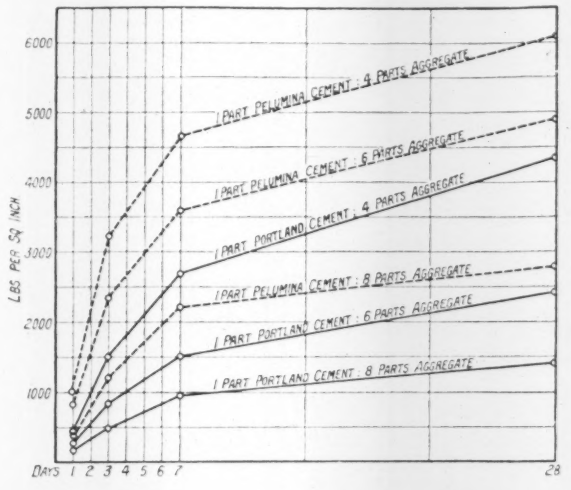

6000
5000
4000
3000
2000
1000
LBS PER SQ INCH
1 PART PELUMINA CEMENT : 4 PARTS AGGREGATE
1 PART PELUMINA CEMENT : 6 PARTS AGGREGATE
1 PART PORTLAND CEMENT : 4 PARTS AGGREGATE
1 PART PELUMINA CEMENT : 8 PARTS AGGREGATE
1 PART PORTLAND CEMENT : 6 PARTS AGGREGATE
1 PART PORTLAND CEMENT : 8 PARTS AGGREGATE
DAYS 1 2 3 4 5 6 7
28

FIG. 2 COMPRESSION TESTS ON CONCRETE

percentage of lime than in portland cement is used with the purpose of increasing the strength at both short and long periods. The hydraulic modulus (the ratio of lime to slag and alumina) is slightly higher than in ordinary portland cement (283 as compared with 266). The silica ratio (the relation between silica and alumina and oxide of iron) is slightly lower (1.84 in the new cement and 2.39 in the portland cement).

The Pelumina cement is, however, not a modification of "ciment fondu" (see MECHANICAL ENGINEERING, vol. 45, no. 9, Sept., 1923, p. 544). The latter sets at the end of 24 hours, while Pelumina cement takes at least 36 hours. Fig. 2 gives data of various compression tests on concrete with and without Pelumina cement. (*The Power Engineer*, vol. 20, no. 229, Apr., 1925, pp. 125–126, 3 figs., *de*)

FOUNDRY

Steel Melting in the Cupola

IN THE Foundry Laboratory at the University of Michigan, Ann Arbor, Mich., there is a cupola with a series of holes cut through the side so that the process of melting can be observed. Experiments were carried on with that cupola to determine how steel melts in a cupola and to compare the melting of steel with that of pig iron and scrap cast iron. To determine the absorption of carbon, pieces of steel were removed from the cupola and studied under the microscope. Gas analysis was made together with other observations. The carbon content in rapidly melted steel was estimated to be between 1.25 and 1.35 per cent, and in no case did it appear to be higher. It seems reasonable that it reaches at least 1 per cent carbon before melting. In melting, steel absorbs carbon, and as the penetration is slight the outside of the piece has a lower melting point than the rest of the piece. The outside melts and runs off, so that steel melts very much as ice does. Pig iron melts in quite another way—somewhat like a piece of brick ice cream in a warm room. The pig iron may be seen to become soft as ice cream softens, and as the softening occurs the outside melts. There are two factors in the melting process, namely, size of stock and the melting point of the material. The temperature in the cupola is so much higher than the melting point of pig iron and steel that the absorption of heat by the piece being melted is of more importance than the abstract melting point of the material. Practical suggestions are made in the article, among other things, that irregular surfaces in steel are desirable. The author believes that it is reasonable to expect heavier pieces of steel to be used as part of the cupola charges than has generally been the case. The oxidation of the steel may be somewhat reduced by using the heavy sizes, and he believes, further, that the averaging of the size of materials is about as essential as the averaging of the chemical composition. (J. Grennan, Instructor in Foundry Practice, University of Michigan, Ann Arbor, Mich., in a paper before the Milwaukee Convention of The Foundrymen's Association. Abstracted through *The Foundry*, vol. 53, no. 8, Apr. 15, 1925, pp. 314–317 and 325, 11 figs., *ep*.)

FUELS AND FIRING

Power Alcohol from Home-Grown Materials in England

A CONSIDERABLE number of experiments have been carried out with a view to determining the possibilities of producing power alcohol from home-grown materials in England. Grain and potatoes appear to be out of the question. Mangolds hold more promise, but are still expensive. The Jerusalem artichoke appears to be the most promising. From experiments made on a semi-technical scale it appears that a yield of about 18.7 British gallons of 95 per cent alcohol is possible per ton of tubers. Furthermore, by a bacillus process butyl alcohol and acetone may be obtained. By a fermentation of the tubers by yeast and the action of an organism belonging to the *bacillus butylicus* a liquor is obtained consisting of 70 per cent of ethyl alcohol, 10 per cent of acetone, and 20 per cent of butyl alcohol. It has been further found that by means of suitable treatment a cellulose can be prepared from artichoke stalks. (Memorandum by Sir Frederick Nathan, published by the Department of Scientific and Industrial Research, 37 pp., abstracted through an editorial in *The Chemical Age*, vol. 12, no. 301, Mar. 21, 1925, p. 265, *g*)

HYDRAULIC ENGINEERING

The Pinard-Sala System of Harnessing Tides

ESSENTIALLY the principle of the Pinard-Sala system is illustrated in Fig. 3 and consists in utilizing the upward push of waves acting in confined spaces. The system consists therefore of a funnel located under water with the wide end open to the incoming waves. The funnel then contracts gradually and ends in a vertical cylinder in which operates a floating piston.

It is claimed that lateral thrust is eliminated. Even with a violent sea the piston rises until the wave has lost its power or has found another way of discharge—which can be provided, of course, by equipping the cylinder in which the floating piston moves with properly located side orifices. The power delivered to the piston is determined, therefore, not by the ratio of the crest of the wave to its trough, but by the total mass forced into the funnel, which is proportional to the difference between the cross-section of the funnel and that of the vertical cylinder. The movement of the piston is of a pulsating character and here is where the first trouble begins, because it is impossible to operate a crankshaft by such irregular oscillating movements. As shown in Fig. 3, the inventors had recourse at first to a ratchet wheel, which had the advantage that it made it possible to utilize not only the upward but

also the downward movement of the piston. Still, even this did not give entirely satisfactory results. Because of this the inventors have completely eliminated mechanical transmission between the water rising in the cylinder and the wheel and have resorted to a construction like that shown in Fig. 4. Here the elbow connecting the horizontal conduit with the vertical cylinder has been eliminated and a conical passage substituted therefor, through which water is discharged against a Pelton wheel.

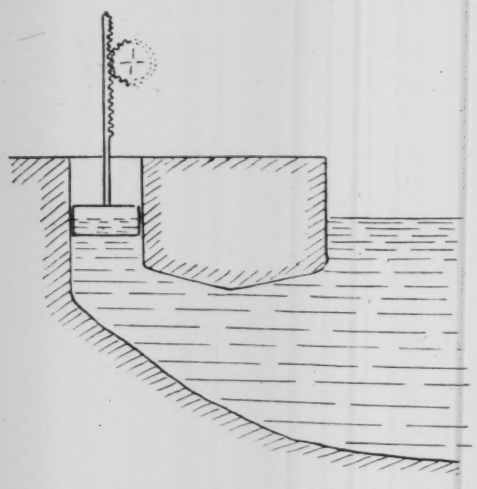


FIG. 3 DIAGRAMMATIC ILLUSTRATION OF THE PRINCIPLE OF THE PINARD-SALA SYSTEM FOR HARNESSING TIDES

In tests made with a conduit of sheet metal 7 meters (23 ft.) long, it was found that with waves which are not of sufficient violence to knock over a man, so much pressure was obtained in the conduit itself and at the wheel that the blades of the latter were twisted in a few moments and the conduit burst.

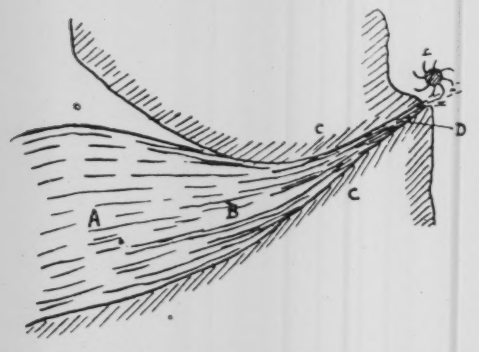


FIG. 4 PINARD-SALA SYSTEM FOR HARNESSING TIDES

Further information, among other things the calculation of power output of a possible installation, will be found in the original article. (F. Pinard and R. Sala, *Bulletin de la Société d'Encouragement pour l'Industrie Nationale*, vol. 124, no. 2, Feb., 1925, pp. 119–134, *d*)

INTERNAL-COMBUSTION ENGINEERING

The Bristol Jupiter Aircraft Engine

THE engine is of interest because while it develops some 400 b.hp. from nine cylinders, it is of the radial type and air-cooled. Its low weight renders it suitable for use in fast machines of the scout type, but it has also given good service in heavier and slower machines carrying freight on the cross-channel services. A three-cylinder engine of the same type as the Jupiter is built by the same company (Bristol Airplane Co.) and is known as the "Lucifer." It develops somewhat above 100 b.hp.

Rated at 380 b.hp. at 1575 r.p.m., the engine develops considerably more on test, the actual power at this speed being from 395 to 405 b.hp. on full load. A standard-production engine recently seen running on a Froude brake developed 405 b.hp. at the normal speed of 1575 r.p.m. and 436 b.hp. at the maximum permissible speed of 1750 r.p.m. The rate of fuel consumption is not at all high, comparing favorably with that of many water-cooled engines. During the same series of tests, nine pints of Castrol "R" were used per hour, this being less than half the permissible quantity and at the rate of 0.022 pint per brake horsepower at the normal speed of 1575 r. p. m.

Complete with the propeller hub and all equipment, the engine weighed 780 lb. dry, which is equal to 2.05 lb. per hp. at rated load. On the test figures quoted above, however, the weight per brake horsepower at normal speed was 1.92 lb., while at maximum speed the weight was only 1.79 lb. When comparing these weights with those of a water-cooled engine, it should be borne in mind that nothing has to be added for radiator, water, piping, or reserve water and tank. (*The Automobile Engineer*, vol. 15, no. 201, Apr., 1925, pp. 96–103, illustrated, *dA*)

Adapting Waste-Heat Reclaimers to Oil Engines

HEAT reclaimed from oil engines may be used for the heating of power plants, workshops, etc. It may also be used for the distillation of liquids, supplying heat for chemical-process work, preheating of boiler feedwater, or maintenance of steam in stand-by boilers of combination power plants, etc.

The article describes a particular case where the requirements were for steam for heating fuel oil so that its viscosity might be reduced for centrifuging, and further for distilled water to make up the evaporation losses of jacket water in a closed circulating system.

In this case it was decided to use a double-effect evaporating and distilling plant. All the steam from the exhaust-gas steam generator was used for the distilling plant, thus producing the maximum quantity of distillate. With the proper arrangement of oil heater it was possible to heat the fuel oil and condense a part of the vapor from the distilling plant.

Calculation would indicate that there are over 2000 B.t.u. per brake horsepower-hour in the exhaust gas. It is claimed that the Davis exhaust-gas steam generator can reclaim 43 per cent of this waste heat, so that 1100 B.t.u. per brake horsepower-hour [Figure cited in the original article.—EDITOR.] can be converted into steam heat, and this in its turn is equivalent to nearly 14 per cent of the total heat used per brake horsepower-hour. It is claimed that with the engines of 2000 b.hp. operating 24 hours a day 300 days in the year with oil at $2.70 a barrel, the saving through waste-heat reclamation would amount to $11,650 per year. (*Oil Engine Power*, vol. 3, no. 3, Mar., 1925, pp. 152–153, 3 figs., *g*)

Beardmore Crude-Oil Aircraft Motor

DESCRIPTION of an aircraft motor under development at the Beardmore shipyards. While the motor has not yet been tried in flight, it has apparently been tested on the ground. The description in this instance is taken from a German publication, because the British authorities are said to have prohibited publication of information about this device in English papers.

The motor works essentially on the Diesel principle with fuel injection, but is also equipped with high-tension ignition. It can be operated with gasoline as well as with Mexican oil of specific weight 0.9, has six cylinders arranged in line, and the cylinders and crankcase form a single large aluminum casting. The camshaft and auxiliaries drive is attached to the rear cover. On one side there are provided three large ports with covers through which the connecting rods and pistons may be taken out, and which also afford access to the crankshaft bearings.

The steel liners for the cylinders are inserted into the cast-aluminum block from above. Each cylinder has a separate cylinder head screwed on to the aluminum block. The camshaft is located entirely within the motor block and is set sideways, and the overhead valves are operated by means of short rods. Each cylinder has four valves radially arranged in the hemispherical combustion chamber. A separate aluminum cap entirely covers the valve-operating mechanism and the spark plugs, which latter are located in the cylinder head between the four valves and are provided for operating the engine on gasoline. When the engine operates on crude oil the spark plugs are removed and in their places are inserted the injection nozzles. The motor is now built in two types, the "Cyclone" having upright, and the "Typhoon" inverted, cylinders. At 1220 r.p.m. the engine, which weighs 820 kg. (1804 lb.), is supposed to deliver 800 hp. (This output is, however, for gasoline as fuel.) The motor will be installed on an airplane at an early date. There is a third type of which still less is known and which contains eight cylinders in line; this motor is rated at 1050 hp. at 1220 r.p.m. and is said to weigh 1070 kg. (2354 lb.). (*Zeitschrift für Flugtechnik und Motorluftschiffahrt*, vol. 16, no. 5, Mar. 14, 1925. Part of a serial article entitled, The State of Construction of Large Aeronautical Motors Abroad, by Alfred Richard Meyl, p. 121, 2 figs., *d*)

The Baldwin Diesel Locomotive

THE locomotive with which experiments are being made at the Baldwin Locomotive Works is equipped with an inverted V-type two-cycle oil engine developed by the Knudsen Motor Corporation and rated at 1000 b.hp. The engine drives electric generators through gears, the wheels of the locomotive being electrically driven. Since the engine has two crankshafts, they are geared together in order that the machine may function as a unit. The increase in speed which is afforded by the gearing will make it possible to employ a generator of higher speed and less weight than would be the case if it were direct-driven.

Two-cycle operation, uniflow-port scavenging, and a Siamese combustion space for each pair of cylinders characterize the Knudsen design. Owing to the fact that one piston of the pair controls

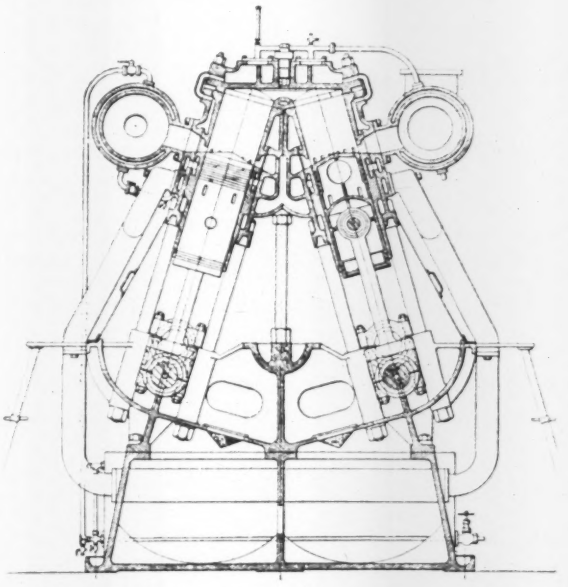


FIG. 5 KNUDSEN ENGINE AS USED ON EXPERIMENTAL BALDWIN LOCOMOTIVE

the inlet scavenging-air ports and the other the exhaust ports, there is a direct sweep of gases through the combination. Several functions are united in the cylinder head: it simultaneously makes oblique joints on both cylinders, accommodates a throat or neck through which air and spent gases pass during the scavenging period, and constitutes the combustion space. The head is supplied through elbow connections with cooling water from the cylinder water jackets, and there is a cored bridge of metal formed between the common combustion space and the obliquely flanged cylinder joints which become nearly tangent in this neighborhood. Sufficient space is reported to be available in the bridge for the accommodation of a core of adequate dimensions, and it is claimed that the circulation of water through the cored hole is sufficient to take the peak of the combustion heat which occurs at this point.

Good scavenging calls for a relatively large passage in the cylinder head, whereas the need for reducing the combustion space between and attaining a suitable compression pressure calls for a reduction of the head space. As the result of inclining the cylinders toward one another in the manner shown it becomes possible to economize space and to make the throat of correspondingly greater area. It would thus appear that the inverted-V construction, in addition to its bearing on the use of dual geared crankshaft, also is conditioned partly by the Siamese combustion space.

In the absence of operating records other than laboratory tests it is difficult to gage at the present time how the new unit will behave when it is placed on locomotive trucks. All that can be said now is that the Baldwin Locomotive officials feel sufficiently encouraged with the progress of the experimental work already made to warrant their fitting the engine to an electric-drive locomotive designed for the purpose. No data of tests have been published. (*Oil Engine Power*, vol. 3, no. 3, Mar., 1925, pp. 159–161, 1 fig., *d*)

The Mechanical Efficiency of an Internal-Combustion Engine

RECENT research work conducted at the Air Ministry Laboratory on a Benz single-cylinder engine to determine (1) whether there was any appreciable difference in the sliding friction when the engine was motored and when under power; (2) whether this was balanced by the difference in pumping losses; and (3) the resultant error, if any, in the assumption that the addition of brake and motoring power gave the indicated power.

The indicated mean effective pressure was found by means of an improved Watson optical indicator. From curves in which the indicated mean effective pressure is plotted against the sum of the brake and motoring mean effective pressures, it appears that the sum of the latter is too low at high loads and too high at low loads. The effect is greatest for diagrams with a sharp explosion curve, in which case the difference amounts to 10 lb. per sq. in. at 140 lb. per sq. in., or 7 per cent. At 120 lb. per sq. in. the error amounts to 4 lb. The pumping losses under power and the total losses when motoring are plotted in two sets of curves. The difference between the total losses and the pumping losses gave a sliding friction equivalent to 11 lb. per sq. in. mean effective pressure when motoring. When motoring with nearly closed throttle the pumping losses were 8.5 lb. per sq. in., falling to 6 lb. per sq. in. at full throttle, whereas under power they were 4 lb. per sq. in. on light load, increasing to 6 lb. per sq. in. at full load. Thus at full throttle there was no measurable difference in the pumping losses when motoring and when under power. At other loads, however, the pumping losses were appreciably less when under power.

An endeavor was made to discover whether the pumping losses could be reduced by a change in the character of the exhaust

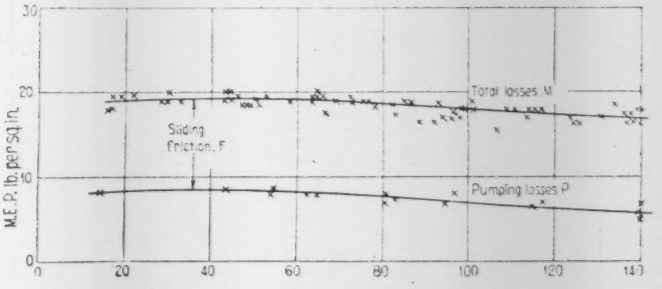


FIG. 6 TOTAL LOSSES AND PUMPING LOSSES WHEN MOTORING

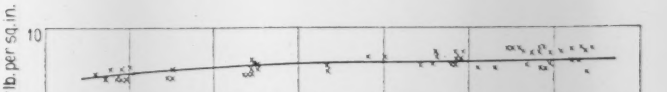


FIG. 7 PUMPING LOSSES UNDER POWER

pipe. The data plotted in Figs. 6 and 7 were obtained with an exhaust pipe consisting of 15 ft. of 3-in. gas piping with a smooth 90-deg. bend at 30 in. from the engine. In these experiments a large expansion chamber was inserted, and it was found that the pumping losses at full throttle were reduced to 5 lb. per sq. in. both when motoring and under power.

The fact that the pumping losses when motoring are usually greater than under power, tends to make the sum of the brake and motoring mean effective pressures more than the true indicated mean effective pressure, and this result is obtained at low loads. At high loads the increase in sliding friction counteracts this effect, and the two balance at about 70 lb. per sq. in. indicated mean effective pressure, while above that pressure the sum of brake and motoring is less than the indicated pressure. As the pumping losses are the same under power and when motoring at full throttle, the error at full load is simply the friction increase.

The determination of the mean effective pressures by the optical indicator was considered to be accurate within 1 lb. per sq. in., while the brake mean effective and total motoring losses were obtained with an accuracy of 0.5 lb. per sq. in. The same experiments were repeated with the R.A.E. electrical indicator; as before, it was found that the sum of brake and motoring pressures is

too high at low loads owing to the difference in pumping losses when motoring and when under power. At high loads the sum of brake and motoring pressures is too low, and as the pumping losses when motoring and when under power are practically identical at throttles giving over 120 lb. per sq. in. indicated mean effective pressure the errors at these indicated pressures is the increase in sliding friction. A considerable increase in friction under load was also found when the spark was advanced.

From tests on various Diesel engines it would appear that the friction losses on full load differ from those at no load, so that the behavior of the Benz engine in these tests is not abnormal. (Dr. H. Moss and W. J. Stern. *The Automobile Engineer*, vol. 15, no. 200, Mar., 1925, pp. 78–82, 10 figs., *et*)

British Marine-Oil-Engine Trials Committee

SECOND report of the Marine Oil Engine Trials Committee, presented in March, 1925, at a joint meeting of the Institution of Mechanical Engineers and Institution of Naval Engineers and dealing with the shore and sea trials of the engines of the twin-screw motor vessel *Dolius* equipped with Scott-Still engines. The first report dealt with the engines of the *Sycamore*, driven by two sets of oil engines of the Richardson-Beardmore-Tosi director-valve type.

The *Dolius* is equipped with two sets of Scott-Still four-cylinder two-stroke-cycle single-acting oil engines. The particular feature of these engines (described in MECHANICAL ENGINEERING, vol. 44, no. 5, May, 1922, p. 316) is that steam generated from waste heat is applied on the under side of the combustion pistons in a single-acting manner. The engines were designed to develop a total of 2500 shaft hp. at 120 revolutions and to propel the vessel fully loaded at an estimated speed of 11 knots.

The report is quite extensive and not well suited to abstracting. From the shore trials it would appear that the fuel consumption per brake horsepower-hour was around 0.360 lb. and the thermal efficiency on a total-indicated-horsepower basis was somewhat in excess of 40 per cent.

In the discussion which followed the presentation of the report, the high mechanical efficiency, which was found to be of the order of 90 per cent, was ascribed to the fact that the engine has neither blower nor air compressor, and that, apart from some small power pumps driven from the engine, the losses associated with the mechanical efficiency were only those produced by the friction of the engine itself. The frictional-horsepower figures showed that the friction losses were equivalent to a mean effective pressure of 7 to 8 lb. per sq. in., a figure confirmed by some special friction tests made after the conclusion of the shore trials.

In the sea trials the fuel consumption per shaft horsepower as determined from torsionmeter readings varied from 0.291 to 0.34 lb. per hour. The discusser, however, did not think that the torsionmeter readings could be accepted as correct.

C. W. J. Taffs, secretary of the Committee, dealt with the discrepancies in the figures arrived at for the torques. Plotting the results logarithmically, he endeavored to show that for the *Sycamore* the various measures of the torque, although they did not agree, were at least consistent in that they all gave straight lines of very nearly the same slope. In the case of the *Dolius*, a similar argument showed that the results were inconsistent among themselves. He suggested that the discrepancies were probably to be accounted for by an error in the calibration of the torsionmeter. The logarithmic plotting also suggested that during the second day's sea trials the torsionmeter was slipping in some manner or other.

E. B. Moullin, the designer of the torsionmeter, said that so far as the calibration of the meter was concerned, it was difficult to discover any possible cause for the discrepancy. A static test had been made on a meter of 400 hp., and the results had been compared with those obtained by the method of calibration used for the meters on the *Sycamore* and *Dolius*. The results agreed to within one-half of 1 per cent. As in the *Sycamore*, the torsionmeter readings were in close agreement with the predicted horsepower deduced by Mr. Baker from model experiments. If the torsionmeter readings were plotted against the torque deduced from the indicated horsepower and mechanical efficiency, a satisfactory straight line was obtained. The line, however, did not go through the origin, and led to the ridiculous conclusion that there would be some torque without any indicated horsepower. This fact might suggest the existence of a zero error in the torsionmeter, but on every occasion on which it was tested the meter came back dead to zero when the run was finished. The maneuvering trials at the end of the first day's tests caused a slight mishap to the meter, which could not be wholly put right before the second day's trials. This fact might have caused the slipping action referred to by Mr. Taffs. G. S. Baker, of the National Physical Laboratory, suggested that the cause of the discrepancies in torque measurements lay with the methods of taking the indicated horsepower. (*The Engineer*, vol. 139, no. 3613, Mar. 27, 1925, pp. 346–349, 3 figs., *e*)

In an editorial in the same issue of *The Engineer*, p. 355, the perplexing discrepancy in the results of tests revealed in the first report is discussed. From the report on the *Sycamore* it would appear that there was apparently a very large loss of power between the engines and a point some distance aft on the propeller shafts. The second report on the *Dolius* is claimed to have made this situation still more obscure, for while on the *Sycamore* all the data indicated a loss of torque, in the *Dolius* tests there had been uniformly a gain of torque. Moreover, just as in the *Sycamore* the loss increased as the power developed was reduced, so in the *Dolius* the gain was greatest at the lowest powers.

It is difficult to believe, according to the editorial, that a scientific instrument designed as is the Moullin torsionmeter on a well-understood electrical principle would behave so erratically as to show a positive error of 56 per cent on one occasion and within a week a negative error of 27 per cent.

MACHINE PARTS (See Special Processes: Drastically Quenched Bolts and Rivets)

MACHINE SHOP

Symposium on Cutting Tools

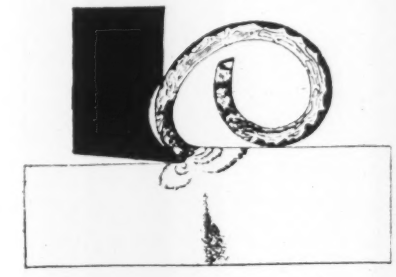
ON MARCH 13, 1925, further reports to the Cutting Tools Research Committee were submitted for discussion before the Institution of Mechanical Engineers. The first, entitled The Action of Cutting Tools, was contributed by Prof. E. G. Coker, and the second, Experiments with Lathe Tools on Fine Cuts and Some Physical Properties of the Tool Steels and Metal Operated Upon, by Dempster Smith and Arthur Leigh.

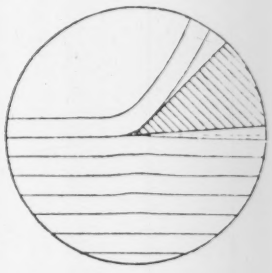
In his paper dealing with action of cutting tools, Professor Coker dealt primarily with the cutting action taking place in milling machines as revealed by his photoelastic method (described in MECHANICAL ENGINEERING, vol. 44, no. 8, Aug., 1922, p. 532). A machine was devised whereby the cutting action of a milling cutter on the edge of a transparent plate of nitrocellulose or other material could be examined and the stresses generated could be inspected by means of the color bands revealed when a polarized beam of light was passed transversely through the work.

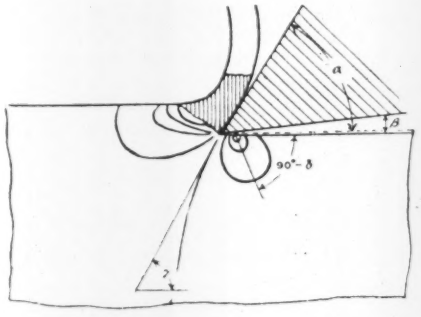
In Fig. 8 a sketch reproduced from a photograph is given showing the action of a planing tool on nitrocellulose. The color bands in the shaving and work and the stress distribution deduced from them present the same essential features as those found in the earlier lathe experiments. The angle γ, Fig 9, of the opaque black brush, stationary in front of the cutting edge, which divides the area under radial tension from the area under radial compression, was found to be greatly different from the cutting angle α of the tool when α was small, but to tend toward equality with α when the cutting angle approached 90 deg. Similarly the inclination δ of the line perpendicular to the axis of the tension lobe tended toward equality with α when α was large, but was widely different from it when it was small. The data derived from these measurements afforded a clue to the general direction of the force exerted by the point of the tool on the work, but it was discovered that the subject was complicated by an "uplift effect" mentioned hereafter.

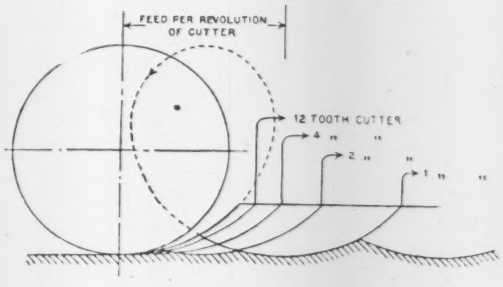
It was observed that after a cut had been taken and the tool run back, the point of the tool sometimes failed to make contact with the work, particularly if the tool angles were such as to leave the surface of the work somewhat rough. The same effect was observed in some cases with metal instead of nitrocellulose work pieces. On a small, new shaping machine it was found that over

a wide range of depth of cut the metal shaving removed was thicker than the depth of cut indicated by the micrometer on the machine. The difference was too great to be accounted for wholly by the plastic stress and strain to which the shaving was subjected. The gap was small when the tool angles used were such as to give clean-cut surfaces, and it was deduced that the gap was primarily a function of the cutting angles. In a new 8-in.-centers Lang lathe, with the tool supported in a most rigid manner, it was found that a tool with a cutting angle of 45 deg. and a clearance angle of 3 deg. when set to a feed of 0.015 in. produced a gap of 0.003 in. with steel, and a still greater gap with nitrocellulose. The action of the tool was observed under a microscope giving a magnification of 40 diameters, and it was discovered that in circumstances producing a gap the tool pulled up the material locally in such a way that—as indicated in Fig. 10—straight parallel lines ruled on the front of the work developed bulges under the tool point. This









FIGS. 8 AND 9 (UPPER LEFT AND LOWER LEFT) PLANING TOOL OPERATING ON A STRIP OF NITROCELLULOSE IN POLARIZED LIGHT

FIG. 10 (UPPER RIGHT) DIAGRAM SHOWING THE DISTORTION UPLIFT OF LINES RULED ON THE WORK PARALLEL TO THE LINE OF MOTION OF THE TOOL

FIG. 11 (LOWER RIGHT) MILLING CUTTERS

distortion diminished regularly in the vertical direction, and tended to vanish at some depth below the cut surface. When this uplift occurs, it is suggested, the consequent tearing action may produce the roughness of the surface of the work noted in such cases. With tools of large cutting angles the peak under the cutting edge may be directed downwardly, while if the tool does not tend to tear the shaving away from the work or force it into the work there is an absence of permanent distortion of the parallel lines ruled on the material. The addition of a trace of lubricant was found greatly to modify the extent of the distortion. It is noted that mechanical operations were often insufficient to insure the absence of all trace of lubricant, and that they had to give way to chemical means.

Passing to the action of milling cutters, the author remarks that the combined rotation and feed cause each tooth to describe an oblate trochoidal curve—see Fig. 11—with the result that the shaving removed is not uniform, but increases in thickness as the cut proceeds. This fact results in features that are not present in cuts of the lathe or planing-machine type. With a single cutting edge, the area in front of the "black brush" springing from the point of the tool is in compression and the part behind it in tension. With a milling cutter, two or more cutting edges are in action simultaneously, and as a consequence at some point in the work between a pair of consecutive cutting edges the pull on the work caused by one edge is neutralized by the push of the other behind it. This fact is revealed under polarized light by the black patch between each pair of teeth. The changes occurring at the two cutting edges do not, however, vary in the same manner at the same time, and as a result it is found that the black patch in the work between the teeth does not remain stationary. In addition, it has been observed that the black brush associated with each cutting edge alters its inclination relatively to the contour of the cut surface as the tooth advances. At first it may be nearly radial to the contour, but toward the end of the engagement it may be 20 deg. in advance of its first position. The point of zero stress indicated by the black patch between the teeth advances toward the cutting edge in front of it as the cut proceeds. The change from compression to tension which occurs at this point is, it is noted, very rapid.

LATHE TOOLS AND FINE CUTS

As regards durability of light cuts, it was shown by E. G. Herbert in 1908 that as the speed is increased the durability rises to a maximum, then falls off, increases to a second maximum, and finally decreases without further interruption. As there was doubt as to the general applicability of these results because they were obtained in a special machine, Dempster Smith and Arthur Leigh determined to carry out a series of tests in an ordinary lathe, using a depth and traverse of cut closely corresponding to those employed in Herbert's experiments. The data of their experimental outfit are given in the original article.

The first series of experiments was directed toward determining the durability of the fine carbon steels as affected by the cutting speed when a very fine cut (0.0625 in. deep and with a traverse of 0.0013 in.) was being taken. Both the hardened and the hardened and tempered tools gave curves exhibiting two maxima. On the assumption that the principal factors affecting the durability of the tool are the resistance to cutting of the work steel and the resistance to abrasion of the tool steel, and on the further assumption that these resistances are measured by the hardnesses of the steels, then, according to the authors, the ratios of the two hardnesses ought to give a curve corresponding to the durability curve when the temperatures appropriate to the cutting speeds are determined. It was found, however, that a curve so constructed only showed approximate agreement, and it was concluded that some other factor, possibly vibration or the work hardness of the tool steel, was affecting the durability.

In the second series of experiments the durability of the high-speed tools was studied as the cutting speed increased and as the cut was altered from extremely light to something approaching a roughing cut. It will be noticed that the cut was increased by keeping the depth constant and augmenting the traverse. On the lightest cut the curve shows two maxima for the durability, namely, at 40 ft. and 155 ft. per min. On the second lightest cut the second maximum is shown at 80 ft. per min., the first being indicated apparently in the neighborhood of zero speed. On the second highest and highest cuts the maxima occur at 55 ft. and 20 ft. per min., respectively. The first maxima have become eliminated. The durability curves thus move toward the left as the cut is increased. The explanation of this phenomenon is stated to be the increase in the temperature of the shaving and work resulting from the increased heat generated by the heavier cuts. It would appear that as the cut is increased the peaks tend to disappear, until when roughing-cut conditions are approached the

curve takes the progressively falling form associated with such roughing cuts.

In the next series of experiments the durability of high-speed tools as affected by secondary treatment and the use of a cooling medium was studied. Among other things it was found that the carbon steel at low speeds has on the average a durability as good as the high-speed steel, from which fact it is suggested that it would be just as satisfactory to employ carbon instead of high-speed steel for files, hand reamers, etc.—tools for light cuts at low speeds. It was also found that at high speeds the durability is not increased by the use of a lubricant and may be decreased. (Abstracted through *The Engineer*, vol. 139, no. 3612, Mar. 20, 1925, pp. 318–320; discussion not abstracted, *epA*)

MARINE ENGINEERING (See also Internal-Combustion Engineering: British Marine-Oil-Engine Trials Committee)

The Kirsten-Boeing Propeller

DESCRIPTION of a new type of propeller developed by F. K. Kirsten, professor of electrical engineering at the University of Washington, working in conjunction with W. E. Boeing of Seattle. Thus far the propeller has been tested only on a 38-ft. motor boat, but it is to be shortly installed on a 75-ft. boat of a type similar to that used by the Government for Coast Guard rum chasers, and is also to be tested on the Navy dirigible *Shenandoah*. The appearance of the propeller is shown in Fig. 12. The mechanism is comparatively simple and consists of a large disk installed on the bottom of the boat and driven by spiral bevel gears from the engine. Extending downward through this disk are several blades which revolve with the disk and also are themselves geared so as to rotate independently in the manner of a sun-and-planet motion.

A condition resulting from these motions (the two motions of the blades on the rotor) is that the plane of each blade (i.e., a plane which passes through the axis and edges of the blade) will pass

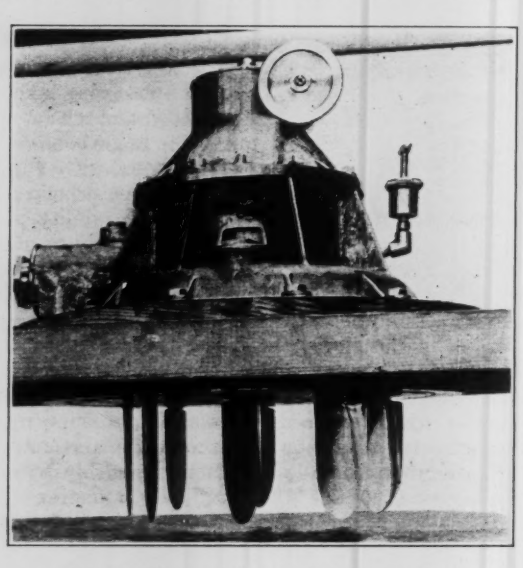


FIG. 12 KIRSTEN-BOEING PROPELLER UNIT

through one point, this point being located in the circular path in which the blades move and which, when set by the controls, is stationary with respect to the motion of the blades.

A second condition which results from the two motions of the blades is that, if a line be drawn through the point on the circular path and also through a point diametrically opposite on the path, the corresponding instantaneous positions of each blade on opposite sides of this line will be symmetrical with respect to it. This line is called the axis of symmetry, and the thrust produced by the blades while being moved as described is always at right angles to this axis and parallel to the plane of the circular path, the direction of the thrust with respect to the sides of the axis being determined by the direction of the rotation of the blades around the center of the circular path.

No rudder or other steering apparatus is required with this propeller as the boat can be steered by shifting the direction of the thrust. The boat can also be driven astern by throwing the thrust forward. The propeller is suitable for twin installations. Tests of the propeller have shown that the boat handles well. On the basis of some tests a propeller efficiency of around 80 per cent is claimed for it.

It would appear that the propeller would have to be rather large to obtain a greater efficiency than the screw for slow-speed vessels, but that it may show a high efficiency for vessels of the wide flat-stern type and fairly high speed. (*Motorship*, vol. 10, no. 4, Apr., 1925, pp. 279–280, 2 figs., *d*)

Relative Commercial Efficiency of Internal-Combustion and Steam Engines for High-Speed Passenger Vessels

ONE large passenger ship has been finished, and others are being completed, with Diesel engines of about 14,000 hp. or more. Some of these are intended to obtain this power on two shafts and some on four. If the former are successful there will be no new experience in applying the same type of machinery to four shafts and so obtaining machinery of 28,000 hp. or more. The engines of one of these ships have been tried at sea, and others that are building have been run successfully for days in the works in which they have been made. It will not be many months before the supreme test of running these high powers at sea will have been made, and if these trials are successful the upper limit of powers of Diesel engines will have been very much extended and the Diesel will have become a direct rival to the steam engine for high-speed passenger ships. Nevertheless the author considers it premature to talk of the Diesel's completely superseding the steam engine, because in the first place the upper limit of power of the Diesel engine is much below the upper limit for the steam engine. [Apparently by "steam engine" the author means both reciprocating and turbine types.] The advantage of the Diesel engine over the steam is that the oil is more efficiently used in the cylinders of the Diesel than in the furnaces of the boilers, but developments are taking place in the boilers and steam turbines which promise to remove some of the differences in economic output, particularly with high-pressure boilers and turbines.

The conclusion to which the author comes is as follows: As far as can be judged from the data available, it seems to be probable that the turbine will give results in commercial efficiency which will make it a strong competitor with the Diesel. It is therefore not the Diesel alone which will make existing steamers obsolete, for developments in the steam engine itself are very likely to produce the same results. The gain in the Diesel over the steam engine is in the smaller consumption of oil per unit of power, but the cost and weight of the Diesel are much greater than the turbine installations using high-pressure steam in the way suggested by Sir Charles Parsons.

The author compares next the performance of the latest large Diesel passenger ship, the *Aorangi*, with that of a steamship of about the same dimensions and power built by the same firm, the Fairfield Company. The steamship is oil-fired with double-reduction turbines. From this comparison it appears that the weights of Diesel engines and steam turbines with tubulous boilers are about the same, while the weight of the high-pressure machinery of the Parsons type would be from 40 to 50 per cent less than either. As regards fuel consumption, including auxiliary machinery, with sea water at 60 deg. and a vacuum of 28 in., the ratio is 0.5 for the Diesel, 1.05 for conventional steam machinery, and 0.7 for Parsons-type high-pressure machinery.

In the case of 20-knot vessels the Diesel type of engine costs £13,000 per year more to run than the steam, while at 17 knots the cost per annum of the Diesel is £7000 more than that for the steam. But the cost of running the steam vessel at 18 knots is only £8000 per year more than for the Diesel at 17 knots. Some investigations have been made for higher power which may be of interest. In the case of the 20-knot vessel with the Parsons type of machinery, it will be seen that a gain of 1052 tons of weight is available in the 3230-mile voyage. Transforming this weight into machinery, fuel, and make-up feed, it is possible to increase the shaft horsepower to 40,000 and the sea speed to 22 knots.

It is to be remarked that the weight of turbine machinery per

shaft horsepower tends to decrease with increase of power. An estimate of an installation of such machinery for 55,000 hp. in a 24-knot ship shows that the weight is about 3600 tons, which is one-third more than that in a 20-knot ship, while the power is more than double. It is not easy to see how a Diesel installation of this power could be applied to a ship at the present time, and its weight would probably be about 8000 tons.

It is, however, in the region of lower powers than those dealt with that the interest of the large majority of shipowners is most centered. It cannot be said that the turbine has displaced the reciprocating engine in this region. The reduction in weight of the Parsons installation is due in part to the use of high-efficiency water-tube boilers and oil fuel, which require higher skill in their handling than the ordinary merchant-ship boiler, coal-fired. But a similar consideration has to be kept in view in the adoption of the Diesel engine in the ordinary type of tramp steamers. Both types involve the training of more highly skilled engine-room staffs, which time only can create.

What is called here the "Parsons type" refers to a proposal by Sir Charles Parsons to build steam turbines working at 500 lb. per sq. in. pressure and 700 deg. fahr. steam temperature, driving propellers through reduction gearing, the steam being generated in water-tube boilers fitted with air preheaters. [Paper read at the World Power Conference in July, 1924.] Other firms were consulted by the writer and are quite ready to guarantee results to equal those promised by Sir Charles Parsons, so that there is solid foundation for the belief in the statement of steam efficiency upon which the comparison with the Diesel engine is based. Further, the Babcock & Wilcox Company has offered equally favorable results in thermal efficiency, first cost, and weight of installation at a pressure of 265 lb. with an installation in which the greater efficiency in the method of producing the steam is expected to make up for the loss of efficiency due to the use of the steam of lower pressure. In this installation coal can be used. Yarrow & Co. designed a Yarrow boiler of 500 lb. pressure and about 700 deg. of superheat for marine engines. Jas. Howden & Co. are building a boiler of the cylindrical and water-tube type combined which will employ 315 lb. pressure and 700 deg. temperature. A damper controls the flow of gases through the superheater. That part of the gases which does not pass the superheater reaches the uptake. The feedwater is introduced in the top back drum, whence it passes down the back rows of tubes to the lower drum and there joins the water coming through the lower pipe from the cylindrical portion of the boiler, and passes upward through the front rows of tubes to the top drum again, from which it enters the cylindrical portion of the boiler at boiler temperature. Both these boilers can be arranged to use oil or coal as fuel. (Sir John Biles in a paper read at the Spring Meeting of the 66th Session of the Institution of Naval Architects, April 1, 1925; abstracted through *Shipbuilding and Shipping Record*, vol. 25, no. 14, Apr. 2, 1925, pp. 392–400, 7 figs., *cdA*)

METALLURGY (See Special Processes: Drastically Quenched Bolts and Rivets)

MOTOR-CAR ENGINEERING

The Supercharged Front-Drive Alvis Racer

DESCRIPTION of a new racing car built in England with a four-cylinder engine 65 by 103 mm. (2.55 by 4.05 in.), equipped with a Roots blower type of supercharger, the engine being capable of developing about 100 hp. on the Froude brake. Complete the car weighs roughly $10^1/_2$ lb. per hp., which is close to airplane figures.

The car is front-wheel-driven by a very simple arrangement. The engine is mounted fairly far back in the frame the wrong way around, i.e., so that the starting handle points toward the rear of the car. At the front of the engine is mounted a bell housing enclosing a special form of a single-plate clutch which couples up in unit a four-speed gear box.

On the extreme end of the gear box, and situated right in the front of the car, is a casing which encloses a spiral bevel reduction gear and a differential, the two differential shafts terminating in brake drums having universal-joint housings central in them. From each universal joint runs outward a propeller shaft terminating in the second universal joint within the hub of each front wheel. These wheels are centrally pivoted for steering, and the universal joints lie on the same center line, so that they are unaffected by the steering action.

Extending forward from the frame there is on each side a twin quarter-elliptic spring, and the ends of these are attached to a twin tubular axis which has its extremities terminating in forks for the steering swivels. Thus it will be seen that while the twin axle, the wheels, and the hubs form unsprung weight, the engine, gear box, and differential casing are all sprung. The back axle of the car consists of a stout tube attached to the frame by single quarter-elliptic springs, and carries bearings for the rear wheels, the latter having brake drums on their hubs. The brakes on the rear wheels are actuated in conjunction with those on each side of the differential casing for the front wheels by pedal or by lever, cable connections being used. Hartford shock absorbers are fitted all around.

The unusual design of the car does not stop at this, however, for the channel-section frame is not made of steel but of duralumin, while the side members are braced together with duralumin tube in steel sockets. This is the first time that duralumin, which combines the light weight of aluminum with the strength of mild steel, has been employed for the entire frame of a British racing car. The driver and his mechanic sit right at the back of the car, within an inch or so of the rear axle; their seating position is so low that it is possible for them to touch the ground with the hand, the actual height of the upholstery of the driver's seat from the ground level being 10 in. (*The Autocar*, vol. 54, no. 1537, April 3, 1925, pp. 599–600, illustrated, *d*)

POWER-PLANT ENGINEERING

The Differential-Pressure Steam-Generating System—Patent Office

THE differential-pressure steam-generating system described in the article is intended for use where the space is very cramped and it is desired to develop a large amount of steam from a small plant. Such a situation exists, for example, on automobiles, trucks and locomotives. It is stated that one unit was installed on a crane at the Brown Hoist Co. The description is not accompanied by drawings and is not clear enough to permit abstracting.

A rather queer story is told about the situation of the patent application in the Patent Office. This is reproduced in the words of the author, an inventor who has to his credit—among other things—the Barlow time airplane bomb used by the American Army in the World War.

"As I say, we are having quite a strenuous patent scrap. I think my greatest difficulty is this—it is a fact, that these things do leak out of the Patent Office. This patent was granted, allowed, and then suddenly there seemed to be great pressure exerted against it, although it was a special patent demanding immediate action, no delays. After it was granted there was suddenly resistance thrown against it, and the patent was withdrawn, and they were citing some twelve different patents which didn't even touch the subject. I took a trip to Washington to answer it, and those were immediately wiped out, but it seemed to me they were after a delay. One thing led to another until the second allowance was made, and then that was withdrawn. The last thing that happened which started a fight that is going to be a trouble maker, was an old, obsolete number in the Patent Office (Under the laws of the country a patent number cannot remain in the Patent Office over two years, that is the Supreme Court decision, without having anything allowed on it and still be a live number). A number considerably older than that was taken and stripped of all data, and the first thing I knew I found a prior claim against me, but all of the matter attached to the claim was identical, every comma and every period, every word exactly as we had written it. When I confronted the Patent Commissioner with that he denied it. I got out the records and proved it. He said it was necessary in determining this contest to copy one side or the other. I don't think the American people intend that at all. That is a situation I don't know where the opposition is coming from, but we are having considerable of it, and that is a little bit of what we are running up against. It is not new, however, I

have known it to happen before." (Lester P. Barlow, Engr., Keith Theater Bldg., Cleveland, Ohio, in *Iron and Steel Engineer*, vol. 2, no. 4, Apr., 1925, pp. 176–184, *dg*)

Nomogram for Calculating Steam Piping from the Thermodynamic Point of View

The author claims that the majority of methods now used for the thermodynamic calculation of steam piping are of the cut-and-try type, while others which give the result directly require long and complicated calculations. He has therefore devised a nomogram (expressed in metric units, however) by means of which many of the problems encountered in the thermodynamic calculation of steam pipes can be easily solved. Among these problems are the determination of such characteristics as diameter, loss of pressure in pipe per meter length, velocity of discharge, output per hour, etc. To determine any of these characteristics it is necessary to know two of them and the state of the steam. The following notation is used:

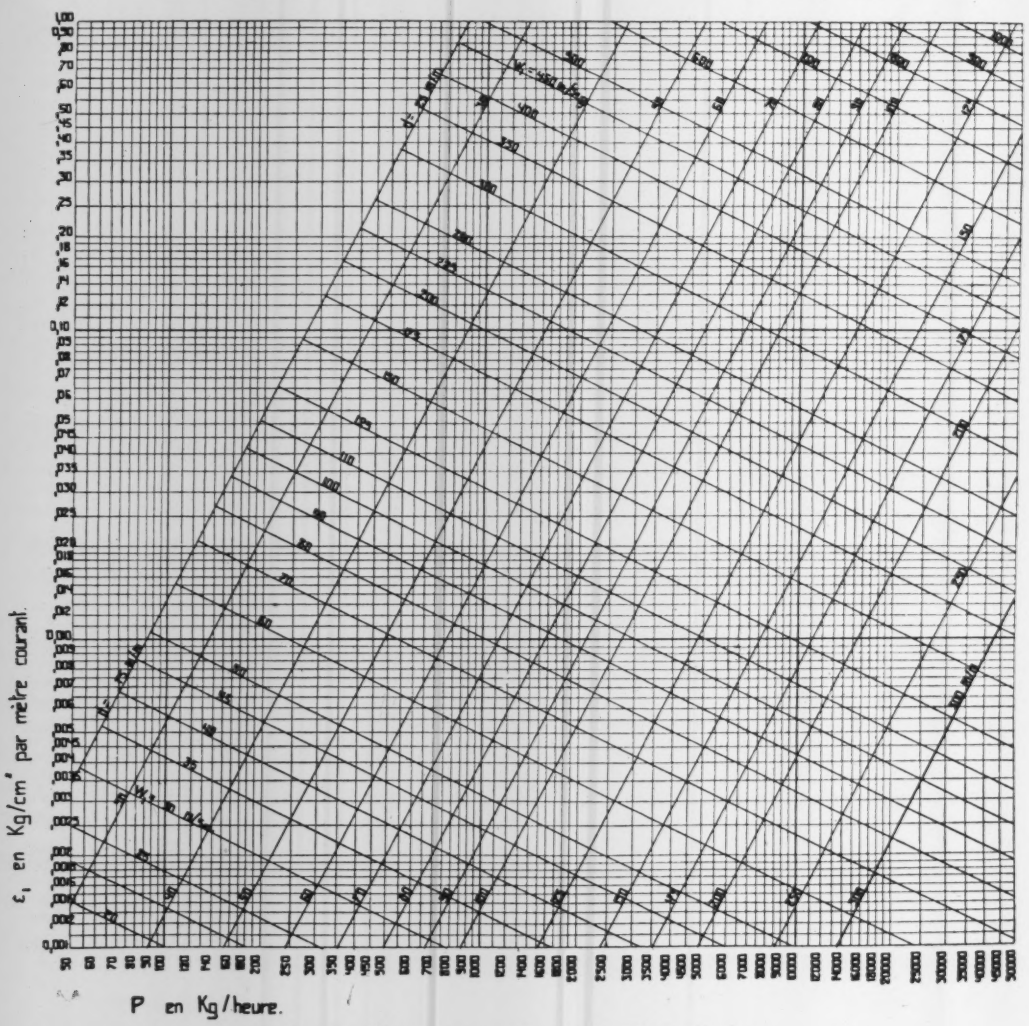


Fig. 13 Nomogram for Computing Steam Piping
(Ordinates ϵ_1 in kg. per sq. meter per meter length of pipe; abscissas P in kg. per hr.)

d = diameter of the pipe in meters
l = length of pipe in meters
ϵ = loss of pressure per meter length of pipe in kg. per sq. cm. per meter (= 4 lb. per sq. in. per ft. of length of pipe)
P = output per hr. in kg. of steam
v = specific volume of steam in cu. meters per kg. (1 cu. meter per kg. = 16 cu. ft. per lb.)
w = velocity of discharge of steam in meters per second
β = a coefficient, the value of which depends on the roughness of the walls of the pipe and the state of the steam
ϵ_1 = loss of pressure for steam of which the specific volume is unity in such piping as used under given conditions
w_1 = velocity of discharge for the same steam
Isometer = line of constant diameters
Isokine = line of constant velocities.

After explaining the formula used in plotting the nomogram reproduced in Fig. 13, the author states that it can be applied only provided the value of β employed is applicable to the given conditions. In this case the author assigned to β the value of 10.55×10^{-8} taken from an investigation by Eberle published in *Zeitschrift des Vereines deutscher Ingenieure*, 1908, as this value would apply in a large majority of the cases encountered in current practice. The nomogram as given is applicable to drawn steel piping for pressures between 3 and 5 kg. abs. (42.66 and 71.1 lb. abs.) and temperatures from saturation to 270 deg. cent. (518 deg. fahr.), or for pressures from 5 to 25 kg. abs. (71.1 to 355.5 lb. per sq. in. abs.) and temperatures from saturation to 200 deg. cent. (392 deg. fahr.) of superheat. In the case of cast-iron pipes the nomogram may be used within the same limits of pressures and temperatures as stated above, by doubling the values of ϵ_1 obtained by calculation.

Method of Employing the Nomogram. In order to show how this nomogram can be employed several examples are cited, one of which follows.

Example 1. It is desired to convey $P = 30{,}000$ kg. per hr. of steam having a specific volume $v = 0.15$ cu. m. per kg. (absolute pressure 20 kg. per sq. m., temperature, 375 deg. cent.). The loss of pressure assumed is 0.03 kg. per sq. cm. per meter length of pipe. What should be the diameter of the pipe and the velocity of discharge for this diameter?

For steam of specific volume v = unity, the loss of pressure would be $\epsilon_1 = \frac{\epsilon}{v} = \frac{0.03}{0.15} = 0.2$ kg. per sq. cm. per meter length of pipe. From the nomogram Fig. 13 it appears that to the point of intersection of the horizontal line passing through $\epsilon_1 = 0.2$ and the vertical line $P = 30{,}000$ there corresponds in the region of "isometers" a diameter giving by interpolation exactly the loss of pressure required, and this is 144 mm. (5.669 in.). In the part showing the "isokines" there corresponds to this point a permissible velocity of $w_1 = 525$ m. per sec., the actual velocity of discharge being then $w = v w_1 = 0.15 \times 525 = 78.6$ m. per sec. In practice a standard diameter of piping would be selected as nearly as possible to the one found from the nomogram. (Auguste C. Raes in *Revue Universelle des Mines*, 7th series, vol. 6, no. 1, Apr. 1, 1925, pp. 31–36, 2 figs. and 1 plate, *tp*)

The Findlay Boiler

The author calls attention to the fact that in ordinary water-tube boilers the velocity of flow of water in the upper tubes is usually lower than in the lower tubes, and that this difference of velocity undesirably affects the efficiency of the plant.

In the Findlay boiler, Fig. 14, in which the tubes used are a combination of water and fire tubes, both banks have the one common outlet series of headers. The mixture of steam and water flowing

up these outlet headers creates a throttling effect at a position immediately below the outlet nipple. The result is that a slightly higher pressure prevails at this point and a slightly lower pressure at the lower portion of the inlet headers. The positive pressure in the top position of the outlet header is said to cause the water in the tube banks to flow down the highest tubes, while the negative pressure in the lower portion of the inlet headers induces a flow in the downward direction. The water tubes in the lower bank are not combined with fire tubes. This is done to insure a plentiful amount of water in the tubes. The tubes are much shorter than usual.

A baffle arrangement is placed in the main steam and water drum. The feedwater on issuing from the internal feed pipe is raised almost instantaneously to boiler temperature, with the result that any sludge or scum is deposited in the feed trough, or in the case of a light scum, is collected at the highest point on the baffle plate. A scum pipe is connected to the drum and arranged so that the scum can be "blown off" by means of the usual valve. The water entering the tube banks is therefore as pure as it is possible to obtain it in boiler practice.

The principal feature of this new design is the inclusion of fire tubes in the usual water tubes in the upper portion of the tube bank. The boiler has a comparatively small amount of water in circulation. It has, however, a reservoir of water in the large mud drum, but this quantity of water is out of the path of circulation and therefore does not absorb heat.

The method of fixing the fire tubes inside of the water tubes is shown in Fig. 15. The products of combustion after sweeping over the water-tube heating surfaces do not pass out to the chimney or economizer but go through the fire tubes. A high velocity of gas flow takes place through these tubes, with the result that the leaving gas temperature must be comparatively low. The boiler requires approximately 20 per cent more draft than the usual type of the same capacity, but this is said to be compensated for by the increase of evaporation. The article states that preliminary experiments have been made on this boiler design. Whether such a boiler has been actually built, and if so, of what size and where, is not stated. (*Engineering and Boiler House Review*, vol. 38, no. 9, Mar., 1925, pp. 372–375 and 376, 2 figs., *d*)

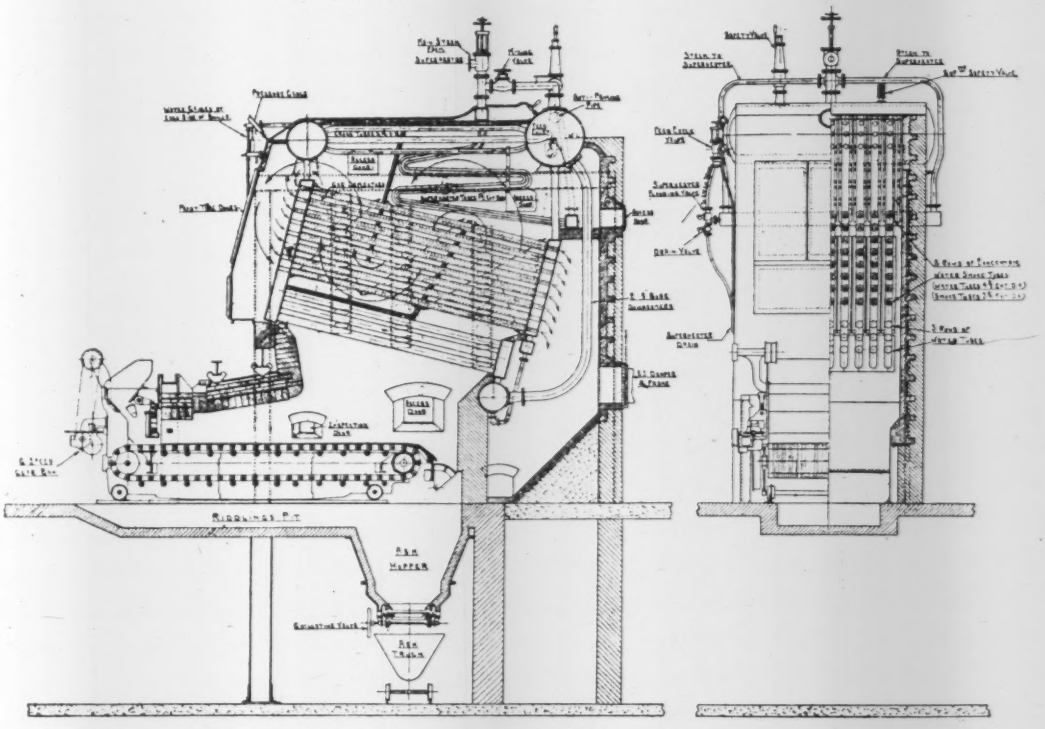


FIG. 14 THE FINDLAY WATER-TUBE SMOKE-TUBE BOILER

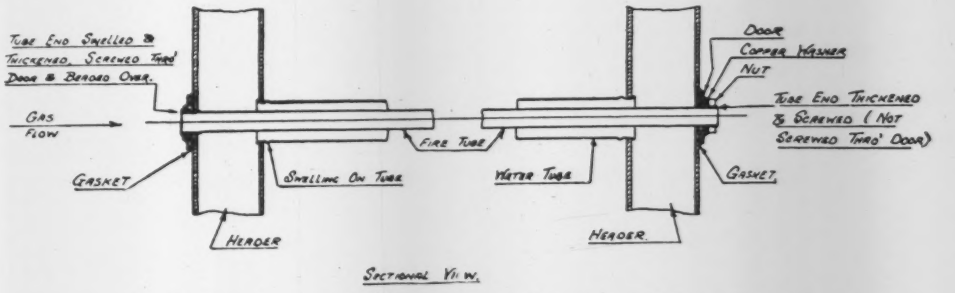


FIG. 15 METHOD OF FIXING TUBES TO HEADERS IN THE FINDLAY BOILER

The Somerset Station on the Taunton River

DESCRIPTION of a power plant built for the Montaup Electric Co. and now nearing completion. The station is located at Somerset, Mass., and the initial installation consists of one 32,000-kw. turbo-generator unit with three 1492-hp. Stirling boilers, this rating being on the basis of 10 sq. ft. of heating surface per hp. Oil is burned now, but provision has been made to burn pulverized fuel.

The boilers are built for a working pressure of 400 lb. They are provided with convection-type superheaters located between the front and middle banks of tubes and designed to give 264 deg. fahr. superheat with 375 lb. steam pressure when the boilers are operating at 200 per cent of rating. Each boiler will be equipped with a single-pass steel-tube economizer having a heating surface of 9220 sq. ft. (the heating surface of each boiler is 14,916 sq. ft.). One boiler has been equipped with a steel-tube-type air preheater and provision is made so that additional air preheaters can be added if desirable.

Both forced and induced draft are used. The forced-draft systems consist of two 30,000-c.f.m. fans for each boiler, each fan being driven by a 40-hp. brush-shifting motor having a speed range of 1048 to 1760 r.p.m. The induced-draft system consists of one 110,000-c.f.m. multivane fan, direct connected to a 150-hp. brush-shifting motor having a speed range of 252 to 814 r.p.m.

The furnaces have been made of large proportions to provide proper conditions for the future use of pulverized coal at high ratings. The combustion chamber, which includes part of the side wall, the bridgewall, and the furnace upper floor and the sides, is ventilated throughout. Since the wall ventilating spaces are subjected to a pressure of $1\frac{1}{2}$ in. under normal operating conditions, the walls are designed to support this pressure, and protection against damage from higher pressure is afforded by the provision of automatic relief doors. Attention has also been directed toward the matter of sealing the walls at all points against leakage to the outside and above the flame zone against leakage into the furnace. Slight leakage into the flame zone of the furnace is not detrimental, as the entering air may be used in combustion and compensated for by a reduction in quantity of air supplied through the burners. The boiler installation is shown in Fig. 16.

Difference in vertical expansion between the part of the wall which is exposed to furnace temperatures and the wall area in the comparatively cool section protected by the tubes is taken care of by a vertical sliding joint opposite the bridgewall, extending up-

ward to intersect with a 2½-in. horizontal expansion space located about 18 in. below the bottom of the steam drums. This vertical sliding joint is ¼ in. wide and is closed on the outside by a 12-in. strip of 1/16-in. asbestos folded lengthwise, the folded edge being inserted into the ¼-in. space about 4 in. and the free edges bent over in opposite directions and pasted to the wall with sodium silicate. The space between the fold is calked with asbestos wicking. The 2½-in. space is closed on the outside by a 9-in. curtain wall built upon a steel bracket attached to the buckstays. A sliding joint is made between the inner and outer wall to allow the inner wall to travel up or down as occasioned by the expansion or contraction of the furnace wall.

The entire boiler-feed supply will be passed through a device which is a combination deaerator and stage heater and takes its

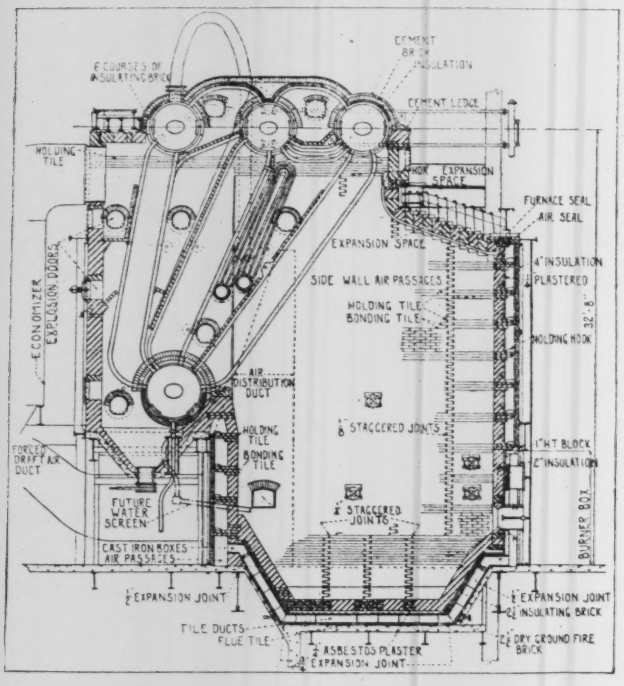


Fig. 16 1492-Hp. Boiler of the Somerset Station, Montaup Electric Co.

normal supply of steam from the twelfth stage of the main turbine. Steam for this purpose may also be taken from the exhaust-steam header automatically through an unloading valve. (*Power Plant Engineering*, vol. 29, no. 7, Apr. 1, 1925, pp. 368–376, 6 figs., *d*)

RAILROAD ENGINEERING (See Internal-Combustion Engines: Baldwin Diesel Locomotive)

SPECIAL PROCESSES

Drastically Quenched Bolts and Rivets

Experiments by Roy H. Smith (Mem. A.S.M.E.), vice-president of the Lamson & Sessions Co., Kent, Ohio, have shown that an extremely high tensile strength—of the order of 160,000 lb., with low elongation, however—can be obtained with steel containing not over 0.15 per cent of carbon by a drastic quench. By "drastic quench" is here meant heating the bolt or rivet to a proper temperature—which should be above the brittleness range, or, say, 1300 deg. fahr.—and then quenching it in water alone or containing soluble oil under conditions of extremely rapid flow of the liquid over the part. To insure unity of results the handling arrangements must be such as to provide (1) that the temperature from which article is quenched is maintained uniform, and (2) that the flow over the articles quenched is also maintained and that none of the articles quenched happens to be in a "pocket" and hence protected from the flow of the liquid. Experiments have also shown that the ultimate tensile strength of the bolts or rivets with the same temperature of quench may be governed entirely by the rate of flow of liquid, being higher when the rate of flow is higher. The arrangements employed are described in the original article. (E. L. Shaner, Assoc-Mem. A.S.M.E., *Iron Trade Rev.*, vol. 76, no. 14, Apr. 2, 1925, pp. 871–874, 6 figs., *d*)

TESTING AND MEASUREMENTS (For Torsion Meters see Internal-Combustion Engineering: British Marine-Oil-Engine Trials Committee)

A New Form of Extensometer

If two equal parallel vertical wires a short distance apart are connected at the middle of their length to a disk which is subjected to a twisting force, a pointer attached to the disk will show a deflection which depends upon the magnitude of the twisting force. If at one end the wires are attached to a horizontal bar that can be moved slightly up or down the deflection of the pointer will be altered when such a movement is made, and if the twisting force and tension on the wires are so adjusted that the original deflection is somewhere about a right angle, a very slight movement of the bar up or down will produce a considerable difference in the reading of the pointer. The magnification thus produced may be very great indeed if the wires are placed sufficiently close together. This, briefly, is the principle of the new instrument, and so far as the writer of the article here abstracted has been able to find out, it represents quite a new departure in methods of measuring small distances.

In applying the method to the construction of an extensometer the chief difficulty lay in finding a method of applying the twisting force without introducing friction. The arrangement adopted is such that a central wire is always exerting a twist upon the parallel fine wires and friction is eliminated.

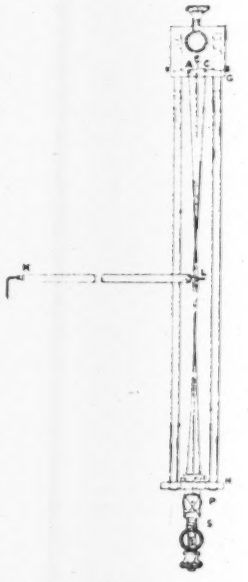


Fig. 17 Chapman Extensometer

As shown in Fig. 17, the two parallel wires *AB* and *CD* really form one wire the ends of which are attached to a brass block at the top, and at the bottom the wire passes over the pulley *P*. These wires pass through the disk *L*, about half-way up, connected to the pointer *K*. The height of the pulley *P*, and consequently the position of the pointer, is adjustable by means of a screw and helical spring *S* at the bottom. The continuous fine wire *ABCD* is of tungsten steel, and has a diameter of 0.0048 in. The central wire *EF*, which supplies the twisting force, is attached top and bottom to a light brass rectangular frame and is of phosphor bronze with a diameter of 0.0201 in. The finer wires are also connected at the top to the same brass frame, but these pass loosely through holes in the bottom of it. The length of the central wire remains constant. The pointer is bent down at the end over the scale, which is of polished aluminum, that acts as a mirror to avoid parallax when taking readings. This particular instrument gives a deflection of 0.4 in. on the scale for an extension of one thousandth of an inch, so that it reads quite easily to the ten-thousandth of an inch with a range of about one twenty-fifth of an inch, but the sensitiveness may be varied within wide limits by setting the parallel fine wires closer or further apart. In this instrument the wires are a quarter of an inch apart.

The deflections of the pointer for a given increase in length of the specimen are not constant throughout the entire range and the instrument has to be calibrated by means of a direct-reading microscope. This is simply done and it is a very convenient instrument to use. Being all in one piece it is easily fixed, the readings are dead beat, and the writer knows of no instrument so simple and convenient with the same power of magnification. (Prof. R. W. Chapman in *Quarterly Bulletin of The Institution of Engineers, Australia*, vol. 2, no. 5, Jan., 1925, pp. 23–26, 2 figs., *d*)

A New Machine for Experiments in Torsion

MOST of the ordinary torsion machines measure the twisting force on the specimen by means of a system of levers. These usually have the disadvantage that they allow of twisting only in one direction, and the accuracy of the balancing depends upon the skill of the operator. In the machine constructed in the testing laboratory of the Adelaide University, the balancing is automatic and the bar may be twisted in either direction so that complete hysteresis diagrams of the behavior of the material may be obtained.

A side and end elevation of the machine are shown in Fig. 18, The twisting force is applied by means of a hand wheel *A* and worm gearing precisely similar to that of the Avery machine. The torque is thus applied to the shaft *B*, carrying at one end a suitable grip for the specimen. The other grip *E* is attached to one end of a standard bar of nickel-chrome steel *C*, which is of such dimensions that it is never strained beyond its proportional limit. One end of this standard bar is rigidly held in the cast-iron frame of the machine and the other end is supported in ball bearings, an overhanging end carrying a steel disk 8 in. in diameter for measuring the angle of twist, and also the grip for the specimen to be tested. With this arrangement the twisting force applied to the test piece is transmitted to the movable end of the standard bar and the angle through which the attached disk turns is a measure of the torque. This movement is recorded by the image of an incandescent-lamp filament on the large circular celluloid scale *J* which surrounds the upper half of the disk and has a radius of 24 in. The deflections of the standard bar under given twisting forces were determined by clamping a lever on to the end of the bar and loading it by direct weight. In this way the celluloid scale was calibrated and the divisions marked to give the direct torque in inch-pounds. In this machine the deflection of the image for the full half-length of the scale corresponds to a torque of 3000 in-lb., the length of the division corresponding to 100 in-lb. ranging from 1 in. at the top to $1^3/_4$ in. at the side, but the range may be varied by changing the standard bar. Two other celluloid scales, *M* and *N*, with lamps *K* and *L*, serve to measure the twist of the specimen from the angular movement of two mirrors clamped on the specimen and 5 in. apart.

The machine has proved a pronounced success in practice and is much preferable to the ordinary type. As the twisting moment on the specimen is increased the image of the lamp filament moves slowly along the scale, the balance being dead beat and automatic. When the yield point of the specimen is reached quite commonly the image moves backward for a time and then slowly goes forward again, a phenomenon difficult to follow by any system of balancing levers. (Prof. R. W. Chapman, University of Adelaide, in *Quarterly Bulletin of The Institution of Engineers, Australia*, vol. 2, no. 5, Jan., 1925, pp. 21–23, 4 figs., *d*)

Gaging at the Point of Manufacture

First of prize winning articles in *Machinery's* contest on interesting gaging devices and methods. The article is not suitable for abstracting generally. Among other things, the author calls attention to the fact that three different principles of construction are used in gaging and that three different human faculties are employed. In the first case, the operation of relieving the tap depends on the eye for gaging. In the second example, the operator depends on holding the work by hand against the stops for the degree of accuracy obtained; while in the last of the three examples the centralization of the work is automatically taken care of.

It is surprising, in the opinion of the author, how often operators are criticized for producing inaccurate work when a simple expedient or a measuring device incorporated in a tool would make it difficult for the operator to machine the work incorrectly. A number of such devices is described in the original article, e.g., a gage that automatically centers a bar in a die for piercing a group of holes. As an example of gaging at the point of manufacture is described the operation of recessing a hole in a sleeve. The provision of dial feeds and tool-locating stop blocks facilitates many operations that must be done accurately and at a reasonable cost, and recommendations as to the design of these devices are made. (Frank H. Mayoh. *Machinery*, vol. 31, no. 8, Apr., 1925, pp. 597–601, 11 figs., *g*)

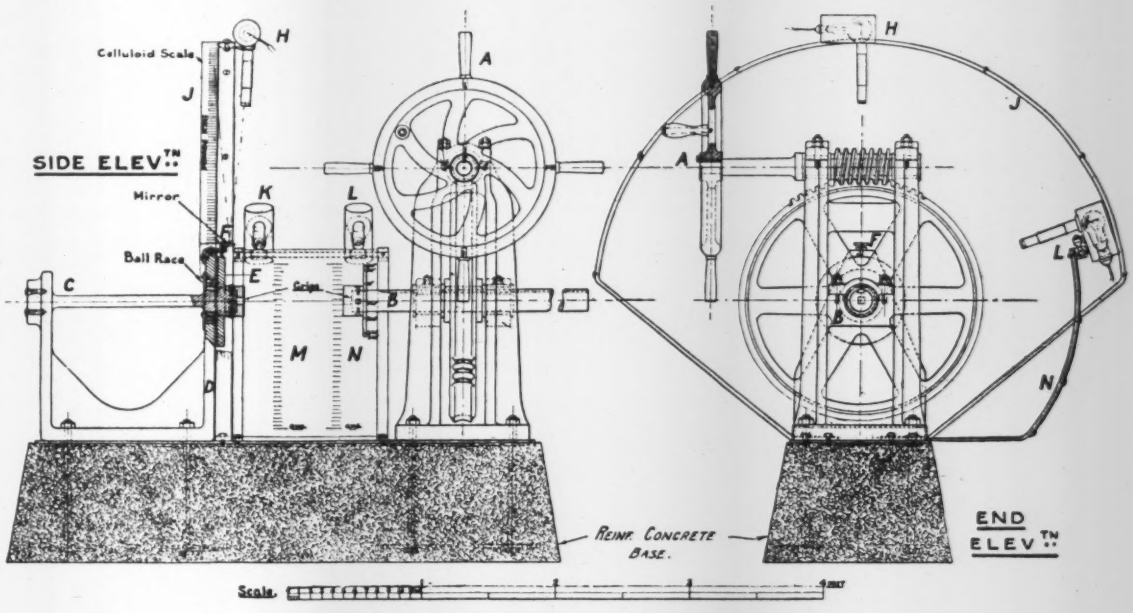


FIG. 18 A NEW MACHINE FOR EXPERIMENTS IN TORSION

VARIA (See Power-Plant Engineering: Patent Office)

FROM data collected by the Federal Trade Commission, it would appear that the average earnings of the big steel companies for the four war years (1915 to 1918), before the payment of bond interest or Federal taxes on income or excess profits, were 7.5 per cent on the investment in 1915, 21.7 per cent in 1916, 28.9 per cent in 1917, and 20.1 per cent in 1918, with a four-year average of 20.2 per cent. Federal taxes took a large part of these profits of course in 1917 and 1918. In this computation the investment is taken as the aggregate amount of stocks, bonds, and surplus. The average earnings of pig-iron makers were somewhat lower, and the same applies to the case of a representative group of beehive-coke companies of Western Pennsylvania (*The Iron Age*, vol. 115, no. 19, May 7, 1925, pp. 1345 and 1390 to 1391, *g*)

CLASSIFICATION OF ARTICLES

Articles appearing in the Survey are classified as *c* comparative; *d* descriptive; *e* experimental; *g* general; *h* historical; *m* mathematical; *p* practical; *s* statistical; *t* theoretical. Articles of especial merit are rated *A* by the reviewer. Opinions expressed are those of the reviewer, not of the Society.

Engineering and Industrial Standardization

Wages and Wastes

THE United States is said to be, by practically every other country, the leading nation of the world. Our natural bump of egotism makes us only too willing to accept a statement of this kind, yet, if we are to be honest with ourselves, we must take issue with it. Other countries with far less opportunities seem to have accomplished even more of some things which are worth while than have we. However, when it comes to murders, robberies, accidents, and other tremendous economic wastes, we unquestionably lead the world.

In the single item of fatal accidents, out of every million of inhabitants we kill each year approximately 860, France 477, Great Britain 452, Japan 446, and Denmark 223.

It is of course impossible to set down in actual dollars and cents the value of a human life, for no one can approximate even the direct loss involved, to say nothing of the indirect loss. Some attempt has nevertheless been made in the state compensation laws, as well as by juries and by various experts, to do so. The figures vary from $5,000 to $10,000 or more; but taking the midway value of $7,500, and the total number of human beings killed in this country each year at 86,000, we have a loss of $645,000,000. If to this be added the further loss of $240,000,000 due to about 2,000,000 workers, worth say, $4 per day, being disabled each year for about 30 days, we have $885,000,000 as the total direct loss. There are, however, other losses involved—indirect ones, which simply cannot be measured, such as the restricted lives of the maimed and their dependents, lessened production, increased overhead expense to society, and so on. These would be even more staggering in amount than the direct loss.

All losses must be paid by the public in some form or other, and result either in increasing the cost of living to all of us, or else in reducing our standard of living.

It will be appreciated that all losses occurring in our plants go into our so-called overhead or indirect expense, and thus are included in the final cost of the product. Were there no competition these losses could be added to the selling price and transmitted immediately to the public, but even that would only help temporarily, since eventually the public would buy less. However, as practically all industries are faced with competition, which limits the increase in prices, losses must be offset by wages, which must be kept at levels sufficiently low to counterbalance.

Let us consider the situation as to wages paid in this and other countries. A comparison made before the war showed that in Germany and other continental European countries they averaged about one-third those paid in this country, while in Great Britain they were about one-half. A few weeks ago I talked with a professor from the University of Charlottenburg, Germany, and he said that wages in Germany now average about one-fifth of what they do in this country.

If now it is impossible to keep foreign-made goods out of this country notwithstanding our high tariff, how much harder will it be for us to compete with countries outside of our own where there is no tariff, or where all goods enter on an equal footing.

It simply means, if we are going to hold up our end and American workmen are to continue to receive their present high wages, enabling them to live on a better scale than foreign workmen, that it is absolutely necessary that losses and wastes of every kind throughout our country must be brought down to the so-called "irreducible minimum." And in this effort the responsibility does not rest upon the employer alone, for wastes are by no means confined to the industries. The responsibility rests also upon the worker and the public generally, for statistics show that as a nation we are inclined to be wasteful and extravagant, taking little thought for the morrow and none for the day after.

There is no more important unsolved problem confronting our country today than that of waste, involving as it does every individual. Unfortunately, the majority of our people do not as yet realize that such a problem even exists, and it is a question whether we have the foresight to awaken in time to the consequences of our present egotism, which keeps us more or less blind to our own shortcomings, or whether we shall have to learn by the bitter and more costly route of experience.

CARL B. AUEL.[1]

Standards for Lumber Should Represent the Best[2]

NEITHER the manufacturer nor the wholesaler nor the retailer is entitled to a profit which does not represent some industrial or commercial service rendered," said Mr. Wilson Compton, Secretary and Manager of the National Lumber Manufacturers Association, in addressing the annual convention of the National-American Wholesale Lumber Association at Atlantic City, March 18, on The Lumber Wholesaler as a Constituent Part of the Lumber Trade.

"These profits and margins will always be unequal, varying with risks, the investments at stake, and the efficiency and skill employed. The point is that they should represent a commercial service. No one is entitled to compensation beyond that. Loose standards of grading and uncertain standards of business conduct interfere with the free, fair, and equal chance to competition for profitable business to which every one engaged in the industry is justly entitled.

"Our standards for lumber and for the conduct of the lumber business should represent the best we can do and not the poorest that we can get away with. National Standards, universally recognized and understood, representing good manufacturing, good grading, and good merchandizing practice, will gradually establish these conditions of fair competition, which it should be our purpose always to maintain."

Revision of the National Electrical Safety Code

THE last edition of the National Electrical Safety Code (Handbook 3 of the Bureau of Standards) was completed in 1920 and issued in 1921. In the following year it was approved by the American Engineering Standards Committee as an American Standard. A revision, preparatory to issuing a new edition, has been under way for the past two years. This work is being carried out by two sectional committees; one dealing with interior work and grounding, and the other with line construction and operating rules.

The latter committee has practically completed its work, and at a meeting at the Bureau on March 5 and 6 final decisions were reached covering requirements which will be contained in Parts 2 and 4 of the next edition of the Code. Among the more important changes in line-construction rules which were agreed upon may be mentioned: Revision of the loading specification upon which the strength requirements of poles, towers, and conductors are based; changes in the map which defines the three loading districts of the country; establishment of a table covering working stresses for steel used in supporting structures; changes in factors of safety for treated and untreated wood poles; changes which will bring about agreement of the Code and the specifications of the American Railway Association; revision of certain clearance requirements; establishment of a new table of flashover values for line insulators; insertion of a rule forbidding the use of the ground as a return circuit in urban districts, and a recommendation against such practice in rural districts; revision of rule covering joint use of poles by supply and signal circuits; and the preparation of special rules governing the construction of radio antennas.

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[1] Manager, Employees' Service Department, Westinghouse Electric and Manufacturing Co., East Pittsburgh, Pa., President of the National Safety Council, and member of the Safety Committee of A.S.M.E.

[2] From the April 7, 1925, issue of *The National Lumber Bulletin*, published by The National Lumber Manufacturers Association, Washington, D. C.

Correspondence

CONTRIBUTIONS to the Correspondence Department of Mechanical Engineering are solicited. Contributions particularly welcomed are discussions of papers published in this journal, brief articles of current interest to mechanical engineers, or comments from members of The American Society of Mechanical Engineers on activities or policies of the Society in Research and Standardization.

Methods of Indicating Tolerance Limits on Drawings

TO THE EDITOR:

In reference to the methods of indicating tolerance limits on drawings described on page 302 of the April issue of MECHANICAL ENGINEERING, Mr. J. W. B. Pearce's method No. 2 is the best, except that the writer believes it should be written

Hole	Shaft
$2.125\ ^{+\ 0.002}_{-\ 0.000}$	$2.123\ ^{+\ 0.000}_{-\ 0.001}$

that is, giving a *minus* tolerance on the shaft instead of a plus one.

The writer would like to quote the following from the Standard Practice for Ordnance Designs.

TOLERANCES

1 When determining the dimensions to prescribe on drawings of fitting members, other than surface fits, the nominal size is the base from which the dimensions are calculated. The nominal size or base is also to be prescribed minimum dimension of the external (female) member. This applies to parts specially machined to size and will not govern when using commercial parts, such as cold-rolled shafting, standard nuts, etc. unless commercial practice so prescribes.

2 The dimension to be prescribed for the maximum internal (male) member is obtained by subtracting from or adding to the base, or minimum internal member, the amount allowed for fit, depending upon whether a clearance or interference fit is desired.

3 The difference between the minimum external member and the maximum internal member will be the minimum allowance for a clearance fit and the maximum allowance for an interference fit.

4 Variations in the size of work are to be tolerated in one direction only.

5 The minimum dimension of the external member and the maximum dimension of the internal member are to be prescribed on drawings. A tolerance of zero value will be applied minus to the prescribed, or minimum, dimension of the external member and plus to the prescribed, or maximum, dimension of the internal member. Tolerances of predetermined value will be applied plus to the prescribed, or minimum dimension of the external (female) member, and minus to the prescribed or maximum dimension of the internal (male) member. Examples:

1 in. nominal size. Clearance fit.
1. Base and minimum external (prescribed on drawing)
0.002 allowance for clearance (min.)
0.998 maximum internal (prescribed on drawing)
0.003 predetermined tolerance.

$1\ ^{+0.003}_{-0.000}$ prescribed dimension and tolerance for external members

$0.998\ ^{+0.000}_{-0.003}$ prescribed dimension and tolerance for internal members

0.002 minimum clearance
0.008 maximum clearance.

Tolerances and allowances may be revised without affecting interchangeability between various shops or countries, so long as the minus tolerance on the standard hole is kept at zero. This is not true of the other methods.

The advantage of this method is that the operator has the dimensions to aim at written in full: 2.125 for the hole and 2.123 for the shaft.

The desired clearance 0.002 is easily perceived, and likewise the limits on the hole (+0.002) and on the shaft (—0.001).

The main advantage of this method and the only one which justifies the existence of any method is that it produces *universal interchangeability*. A universal or common "zero line" or "interference line" is established. This is the minimum size of the hole.

In reference to the method suggested by Mr. Widell, it is believed that with tolerances written in the manner suggested above it is *plainly indicated* just which dimension is wanted. The dimension which should be wanted is the minimum hole and the maximum shaft, which are written in full. Other hole and shaft sizes are "tolerated" for reasons of economy.

T. F. GITHENS.[1]

Cleveland, Ohio.

Properties of Matter under High Pressure

TO THE EDITOR:

In the very interesting paper recently presented by Prof. P. W. Bridgman, entitled Properties of Matter under High Pressure,[2] the experiments dealing with the fracture of materials seem to contradict well-known strength theories. Such strength theories are of great practical importance, and in so far as ductile materials are concerned the writer feels that the experiments mentioned do not rule out the accepted theory within the limits of usual application.

Validity of Shear Theory. It must be noted in the first place that a single theory does not exist which can be considered as satisfactory for all cases and which can be applied for any kind of material and for any type of fracture. Usually one considers two types of fractures: (*a*) fracture due to separation of the particles across some plane perpendicular to one of the principal stresses, and (*b*) fracture due to sliding. It is only for this second type of fracture that a satisfactory strength theory exists which is in good agreement with known experiments. This theory, developed by Mohr,[3] is based on the assumption that of all the planes going through any point of a body under stress and subjected to the same normal stress, the weakest is that in which the greatest shearing stress takes place. On this assumption the strength will depend on the ratio between the maximum and minimum principal stresses at the point under consideration. The third principal stress has no effect on the strength. In practical applications this theory is used only for predetermination of the loading at which the yielding of the material at a dangerous point in the structure begins.

For such ductile material as structural steel the yield points for simple tension and for simple compression may be taken as equal and it may be assumed without substantial error that in all cases where the maximum and minimum stresses have opposite signs (one being tension and the other compression), Mohr's theory coincides with the maximum-shear theory. From this it follows that in such important cases as tension, compression, bending, twist, and combined bending and twist of prismatic bars, the simple maximum-shear theory can be used.

In the more complicated cases of three-dimensional-stress conditions in which all three principal stresses have the same sign, a generalized form of Mohr's theory of strength must be used. It has been experimentally shown[4] that if a hydrostatic pressure from all sides is superimposed on a simple tension the limiting shearing stress at which the sliding begins increases. If all three principal stresses are tension, the sliding may begin at very small shearing-stress values and the fracture of a ductile material under such conditions does not show a pronounced permanent set. Examples of such types of fracture are evident in cases of fracture produced by residual stresses or by stresses due to non-uniform heating. In cases of internal cracks or flaws in forgings or castings of ductile material a breaking may occur suddenly, accompanied by an explosive report and without a pronounced permanent set.

Cylinders under Internal Pressure. In Professor Bridgman's experiments with cylinders under internal pressure it would be expected

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[1] M.E., Engrg. Dept., Cleveland Twist Drill Co. Mem. A.S.M.E.

[2] Lecture delivered at the Annual Meeting of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, December 4, 1924. Published in MECHANICAL ENGINEERING, March, 1925, pp. 161–169.

[3] *Zeit. Ver. deutsch. Ing.*, 1900, p. 1524.

[4] T. Kármán, *Zeit. Ver. deutsch. Ing.*, 1911, p. 1749.

that the yielding of the material would start at the inner surface. But from this it is not justifiable to conclude that the rupture will begin also at the inner surface. Beyond the elastic limit the stress distribution across the thickness of the wall of the cylinder will not follow the simple law given by Lamé's equation and can be represented accurately enough by using the theory of plastic deformation.[1]

Assuming the cylinder to be plastic throughout, the principal stresses in the radial and tangential directions[2] will be represented by the following equations (see Fig. 1):

$$R_1 = 2\,S \log_e(r/b) \quad [1]$$

$$R_2 = 2\,S\,[1 + \log_e(r/b)] \quad [2]$$

$$P = 2\,S \log(b/a) \quad [3]$$

in which

R_1 = principal stress in a radial direction (Fig. 1)
R_2 = principal stress in a tangential direction
S = limiting shearing stress at which sliding begins
P = internal hydrostatic pressure
b = outer radius of the cylinder
a = inner radius of the cylinder
r = distance of the point under consideration from the axis of the cylinder.

Taking $b/a = 7$, for example,

(*a*) At the inner surface of the cylinder:

$$R_1 = -P = -2\,S \log_e 7 = -1.95 \times 2\,S$$
$$R_2 = 2\,S\,(1 - 1.95) = -0.95 \times 2\,S$$

(*b*) At the outer surface of the cylinder:

$$R_1 = 0$$
$$R_2 = 2\,S$$

Comparing (*a*) and (*b*) and assuming the usual type of enveloping curve for limiting circles of Mohr's theory, it will be expected that the rupture will start at the outer surface. The sliding must run toward the center approximately as an equiangular spiral. This is in good agreement with the Bridgman's experiments and some earlier ones.[3]

It is interesting to note that when a sudden bursting of a cylinder occurs, the inner part in any case shows sliding. This type of fracture is very similar to that of a tensile-test fracture of mild steel. The inner part of the specimen usually shows the fracture across some plane perpendicular to the axis of the specimen while the outer part shows sliding.

True Elongation in Testing. The enormous stretching of the inner surface without rupture as obtained by Professor Bridgman in some of his experiments does not represent a striking feature and is of the same magnitude as is usually obtained during a simple tensile test. In order to show this, it is only necessary to keep in mind that during a tensile test the plastic deformation is concentrated principally at the cross-section of fracture and the unit elongation figured on the basis of a 2-in. gage length has no physical meaning. In order to obtain the maximum elongation at the cross-section of the fracture, the reduction of area of this cross-section must be measured. Let A_0 denote the initial area of the cross-section and A the area after fracture. Then, assuming that the density of material remains unchanged, we have the following expression for the true elongation at fracture in per cent:

$$e = [(A_0/A) - 1] \times 100 \quad [4]$$

For instance, in the case of mild steel with 36 per cent elongation and 70 per cent reduction of area, the actual elongation at fracture is

$$e = \left(\frac{1}{1 - 0.70} - 1\right) \times 100 = 233 \text{ per cent}$$

that is, 6.5 times larger than is given by usual method of calculation.

In the case of annealed copper[1] with 54 per cent elongation and 75 per cent lateral contraction, the true elongation is

$$e = \left(\frac{1}{1 - 0.75} - 1\right) \times 100 = 300 \text{ per cent}$$

that is, 5.5 times larger than that given by usual calculations.

The elongations obtained on the basis of Equation [4] are of the same magnitude as those obtained by Professor Bridgman in his experiments.

It is interesting to note that the elongations which we obtain in the usual routine bending test also become many times larger than those calculated in the usual way from a tensile test.

The value of the elongation at the exterior surface, where rupture actually does occur, must depend not only on the tangential stress but also on the longitudinal tensile stress, i.e., on the ratio between the outer and inner radii of the cylinder. In the case of thick cylinders it can be expected that when breaking occurs suddenly the value of elongation at the exterior surface will be comparatively small.

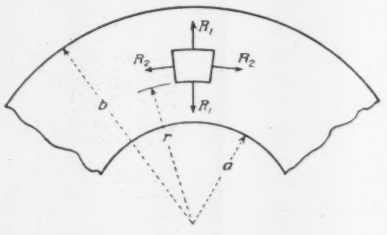

FIG. 1

The "Pinching-Off Effect." Rupture of ductile materials by the "pinching-off effect" represents a very interesting feature. Tests of this sort do not seem to have been made hitherto. The results obtained can be put in agreement with Mohr's theory. It is only necessary to assume that shearing stress necessary to produce fracture at compression must be larger than in the case of tension.

The coincidence of the yield point and the rupture point in the case of the pinching-off effect does not depend on the properties of the material but on the arrangement of the experiment. It is easy to see that the necking of the specimen in these experiments brings about an increase in the longitudinal tensile force, while in the usual tensile test the necking partially releases the specimen.

The pinching-off effect in the case of brittle material is in good agreement with the previous experiments of the same kind made by A. Föppl. He showed[2] that in the case of concrete the maximum shear necessary to produce fracture when the principal stresses consist of a compression equal to the hydrostatic pressure on all planes including the axis is the same as in the case of simple compression.

Cylinders Subject to External Pressure. In discussing the experiments with cylinders subjected to external pressure Professor Bridgman makes a statement with which the writer cannot agree. He thinks that the collapsing of the cylinder subjected to external pressure depends "on the very slight departure from perfect geometrical symmetry. That there is collapse at all must evidently be due to some slight geometric imperfection." It is known that the collapsing within the elastic limit depends on the dimensions of the tube and the value of modulus of elasticity only. A geometrically perfect tube will collapse when the external pressure attains a certain *critical* limit, and a geometrically imperfect tube will sustain the pressure if this is below the critical value.

Large Hysteresis Effects. The unusually large hysteresis effect obtained in the experiments with cylinders submitted to cycles of external pressure may be explained in the following manner. During application of pressure the yielding of steel at the inner surface is produced and permanent sliding occurs. Now, during removal of pressure the outer part of the cylinder, working elastically, develops forces which are sufficient to produce in the inner part of the cylinder plastic sliding in an opposite direction. The area of hysteresis will then represent not the property of material but the property of the structure itself, and in the case under consideration will depend on the ratio between the outer and inner radii of the cylinder.

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[1] See abstract of St. Venant's paper in Todhunter and Pearson's History of Elasticity, vol. 2, art. 261.

[2] From the investigation by L. B. Turner [see Cambridge Phil. Soc. Trans., vol. xxi (1913), p. 388] it follows that the longitudinal stress is never large enough to be a component of the plastic shear.

[3] See W. Kruger, *Mitteilungen uber Forschungsarbeiten*, vol. 87 (1910); and A. Nadai, *Zeitschrift fur Physik*, vol. 30, p. 113 (1924).

[1] Experimental data taken from article by P. Ludwik in *Zeit. Ver. deutsch Ing.*, vol. 68, p. 212 (1924).

[2] Föppl, *Mitteilungen aus. d. Mech. Tech. Labor.*, Munich, 1910.

Excessively high stresses obtained without fracturing in capillary glass tubes represent a very interesting feature. In connection with this a very important paper by A. A. Griffith may be mentioned.[1] In his work with glass Griffith succeeded in obtaining without fracture such stresses as 900,000 lb. per sq. in. A very important theory of strength of a strip weakened by a crack is developed in the same paper. The possibility of a spreading of the crack is considered from the point of view of changes in the complete energy of the system.

In conclusion, it may be said that Professor Bridgman's experiments show some very interesting features which may be of importance in developing strength theories, but they do not dispose of the maximum-shear theory as the criterion of yield for such important cases as tension, compression, bending, and twist of bars of ductile material.

S. TIMOSHENKO.[2]

East Pittsburgh, Pa.

A.S.M.E. Boiler Code Committee Work

THE Boiler Code Committee meets monthly for the purpose of considering communications relative to the Boiler Code. Any one desiring information as to the application of the Code is requested to communicate with the Secretary of the Committee, Mr. C. W. Obert, 29 West 39th St., New York, N. Y.

The procedure of the Committee in handling the cases is as follows: All inquiries must be in written form before they are accepted for consideration. Copies are sent by the Secretary of the Committee to all of the members of the Committee. The interpretation, in the form of a reply, is then prepared by the Committee and passed upon at a regular meeting of the Committee. This interpretation is later submitted to the Council of the Society, for approval, after which it is issued to the inquirer and simultaneously published in MECHANICAL ENGINEERING.

Below are given interpretations of the Committee in Cases Nos. 478, 487, 488, 489, and 490, inclusive, as formulated at the meeting of March 20, 1925, all having been approved by the Council. In accordance with established practice, names of inquirers have been omitted.

CASE NO. 478

Inquiry: What stresses are allowable on through stays welded into the heads or sheets of steel heating boilers? Attention is called to the fact that the Heating Boiler Section of the Code provides only for welded staybolts in Par. H-21.

Reply: It is the opinion of the Committee that where through stays are welded in the heads or plates of steel heating boilers in accordance with the requirements of Par. H-83 and are of a length between supports exceeding 120 diameters, the maximum stress given in item *c* of Table H-4 is allowable provided the stay is supported at intervals not exceeding 6 ft., and a value of $C = 135$ may be used in the formula in Par. H-21.

CASE NO. 487

Inquiry: Is it permissible, under the requirement of Par. U-62, which specifies that in the case of vertical cylindrical vessels subject to corrosion the bottom head, if dished, must have the pressure on the concave side to insure complete drainage, to use what is known in the trade as a "double dished head," namely, a dished head with a reversed flange so that both the dish and the flanges are directed outwardly or toward the end of the vessel?

Reply: Where the vessels are located permanently in a vertical position, it is the opinion of the Committee that the use of the form of construction described does not provide or allow complete drainage and therefore does not comply with Par. U-62 of the Code for Unfired Pressure Vessels.

CASE NO. 488

Inquiry: Is it the intent of Par. P-917 of the Code, that the corner radius of dished heads should be limited to a maximum, when it is understood that the larger the corner radius of a dished head, the stronger it will be, up to the ultimate condition of hemispherical form? It is pointed out that in the new Code for Unfired Pressure Vessels there is no maximum limit for corner radius of dished heads.

Reply: It is the opinion of the Committee that the upper limit of 4 in. for the corner radius of an unstayed dished head given in Par. P-197 of the Code is inconsistent, and accordingly, this matter will be proposed for revision and addenda to the Code. It is the opinion of the Committee that if it is found necessary to use a longer corner radius than that referred to, there should be no objection to it.

CASE NO. 489

Inquiry: Is not the second section of Par. P-194 of the Code an unnecessary duplication of the last sentence of the first section thereof?

Reply: The last sentence of the first section of Par. P-194 was inserted in the 1924 Edition of the Code as a revision of the requirement which appears in the second section of this paragraph, and the latter should be omitted. The Committee therefore calls attention to this typographical error and recommends that the second section of the paragraph be disregarded.

CASE NO. 490

Inquiry: Is there not an error in the discharge capacity given in Table P-11 of the Code under the $^3/_4$-in. outlet at 75 lb. gage pressure, which is given at 198 lb. of steam per hour?

Reply: The discharge capacity of 198 lb. of steam per hour given in Table P-11 for the $^3/_4$-in. outlet at 75 lb. gage pressure is an error of calculation. The Committee points out the fact that a careful checking of the calculations of this table indicates that this figure should appear as 223 instead of 198, and the Committee therefore calls attention to this typographical error.

Suggested Rules for the Care of Power Boilers

THE table of contents of this proposed new section of the Boiler Code, together with the first three sets of rules and part of the fourth set of this section, was published in MECHANICAL ENGINEERING for May, 1925, pages 371–378. Copies of the complete Code may be obtained upon application to the Secretary of the Boiler Code Committee, 29 West 39th St., New York, N. Y. The remaining sets of rules of this section will not appear in MECHANICAL ENGINEERING as the section has been presented at Public Hearing at the Spring Meeting of the Society in Milwaukee and is soon to be issued as a completed section of the Boiler Code.

Plastic Refractories

CLAY refractories, furnished to the trade in their plastic form, are rapidly supplanting special shapes in boiler settings and similar installations. The demand for this material for repair work is also rapidly increasing, so that at the present time it has been estimated that the total daily output exceeds 100 tons. Because of the increasing importance of this commodity to the trade, and the fact that the Government is contemplating the formation of specifications for its purchase, the Bureau of Standards undertook to determine the physical and chemical characteristics of several commercial brands of these plastic refractories.

Test specimens, made by hand-ramming in molds, were burned, subjected to standard and modified tests for fireclay refractories, and also analyzed chemically and petrographically. The information obtained indicates that plastic refractories as now furnished to the trade are inherently the equivalent of the highest-grade fireclay brick. A comparison of softening-point determinations and chemical analyses by three independent laboratories shows that a variation of from one-half to one cone in the former, and more than 5 per cent on some constituents in chemical analysis, can be expected for different samples representing one brand or product.

[1] See Phil. Trans., vol. 221, p. 163–198 (1920); also British Assn. Adv. S. (1921).

[2] Engineer, Westinghouse Research Laboratory. Mem. A.S.M.E.

MECHANICAL ENGINEERING

A Monthly Journal Containing a Review of Progress and Attainments in Mechanical Engineering and Related Fields, The Engineering Index (of current engineering literature), together with a Summary of the Activities, Papers and Proceedings of

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What is an Engineer and How Much Pay Should He Receive?

IN 1921 the International Labor Office[1] at Geneva undertook an inquiry into the comparative status of "technicians" in different countries. It was soon found that the term "technician" was meaningless and the inquiry was abandoned. Later it was again taken up but the field was narrowed to include only chemists and engineers, and the result has been issued in the form of a ninety-one page pamphlet. The inquiry and findings are under five headings, namely,

I The Title of Engineer—How Obtained and to What Extent is it Legally Protected
II Employment—Methods of Finding Employment—Migration, etc.
III Conditions of Employment—Hours of Work—Remuneration, etc.
IV Insurance and Provident Institutions
V Organizations and Demands of the Profession.

The number of countries investigated was sixty-four. The method of making the inquiry should be noted and carefully weighed in accepting or rejecting the conclusions reached by the investigation. A limited number of carefully selected individuals in each country were requested to answer the necessary questionnaires instead of sending out the inquiries broadcast which is the usual method. Whether such a method of securing information is sound may be questioned, since if the number of correspondents is small the answers partake of the nature of personal opinions rather than that of a statistical study.

As might be expected, the inquiry as to what the title of "engineer" should signify was not productive of any definite conclusion. Perhaps no professional title has been so much abused as that of engineer. It has, in fact, lost much of its significance in this country. The Report states that "it is extremely difficult to determine what is the distinctive work of an engineer and what conditions must be attached to the right to use the title. Some countries have endeavored to solve the problem; in others it is untouched; while in others, again, it does not appear as yet to have become acute." This is an interesting statement in view of the growing tendency in this country to license engineers. Clearly this cannot be done intelligently unless specific definitions of the content of the profession can be drawn. The Report brings out a wide divergence of opinion on this point and indicates a wide variance of practice in different countries.

Section II, on employment, is important only in that it gives a digest of the degree of unemployment of engineers at the time the inquiry was made, and is probably out of date already.

Section III is the most interesting part of the Report, since it raises the perennial question as to the remuneration of engineers and gives a casual survey, at least, of the financial status of the engineer and chemist in a large number of countries. The Report frankly admits the difficulty of accurate classification of engineers and the impossibility of making comparative statements between different countries. What it has to say regarding the United States is, however, of interest. In this regard I should like to refer to another document bearing upon this point.

The Editor of the Journal in transmitting the Report to me for review accompanied it with the following quotation from an article by Mr. Fred Kelly in *The Nation's Business* for December, 1924.

> I sat one evening listening to two fairly successful engineers one of whom is now a business executive, discussing the present state of the engineering profession. They agreed that a competent engineer is likely to be found working for a plain business man who is not an engineer.
>
> "I can hire a fairly good engineer," declared one, "for less money than I'm obliged to pay a first-class stenographer. Of course, such an engineer won't have much imagination or initiative, but he will do what he's told accurately and acceptably."
>
> The other engineer remarked that he is sending his two boys to a college where they will get an old-fashioned cultural course—mostly Latin and Greek—largely for the purpose of diverting them from a line of study which might lead to a degree in engineering.
>
> Both declared that an engineer will not ordinarily go far because he is too busy doing one routine thing which can easily be done by another younger engineer whenever he gets tired of it.
>
> I was astonished that engineering could be held in such low esteem by those within the profession. Is the trouble that there are too many engineers? Certainly it isn't that they are lacking in fundamental intelligence. Army tests applied to officers during the World War showed that engineers were not only at the top in the intelligence scale, but 60 per cent ahead of medical officers at the bottom of the scale. I wish other business men and engineers would write and tell me their impressions of opportunities or lack of opportunities in engineering work at the present time.

Of course such statements make excellent material for popular magazines, but no well-informed man would take them seriously. They are in fact paraphrases of what any one will hear in the inner circles of any profession. If the words, "physician," "teacher," "preacher," "chemist," be substituted for "engineer" in this document it will apply equally well. Each and all of these professions believe that they are greatly underpaid, and perhaps they are, but there is one phase of the discussion that should not be overlooked. All trades, callings, and professions require many men of varying ability. It is well that this is so, since all of us cannot become leaders and masters in our chosen calling. If the man who is quoted in the foregoing wishes to secure a minister to perform some elementary ceremony like marrying him or preaching his funeral sermon, he can find thousands of men fully capable of doing the job. But if he wishes to find a man to fill the pulpit of a great metropolitan church he will find his choice limited to a score of men. If he wishes to be vaccinated he will find hundreds of doctors willing and capable of doing the work for a pittance, but if he must have an operation to remove a defective kidney he must pay the price of a highly skilled surgeon. So if he wants a wheelbarrow designed and built he will find plenty of engineers, so-called, who can produce a first-class wheelbarrow; but if he wishes to build a great bridge or a steamship like the *Leviathan* or have built for him a 50,000-hp. turbine, the salary he pays his stenographer will not go very far in these directions.

Opinions are all right in their way, but they do not always agree

[1] An important part of the scheme for the League of Nations provided for in the Peace Treaty of Versailles, and which involves the meeting of representatives of all states in the League annually in conference for the purpose of drawing up international agreements for regulating and improving industrial conditions. Founded on the belief that there must be international coöperation among the nations if suicidal competition, leading to much human misery and possibly to future wars, is to be avoided. The office maintains a staff of economists and issues numerous publications, the *International Labour Review*, published monthly in English in London, being its most important periodical.

with the results of statistical facts. In the year 1924 there were 20,615 mercantile failures in the United States, exclusive of bank failures, involving a loss of $543,225,449. An analysis of the causes of these failures shows that the larger portion of them were due to ignorance and lack of business ability on the part of those conducting them. There does not appear to be any such corresponding record of failures of engineering structures due to the ignorance of those designing and erecting them, and if there were, technical schools would soon be deserted and the engineering profession would disappear, since no one would trust those who follow it.

Let us be fair and honest in our appraisal of all professional activities. The business man is far from being infallible: as a matter of fact and record he is often densely ignorant of important features of his calling. Accurate costs, for instance, are quite new in manufacturing and still unknown to many business men who freely criticise the professions. Incidentally the man quoted in the foregoing, who deliberately sends his boys to a certain kind of school "largely for the purpose of diverting them from a line of study which might lead to a degree in engineering," shows the same dense ignorance of education and of modern educational thought. It may happen that these boys should normally go to the type of school the father has selected, but to deliberately divert a boy away from his natural bent is to court educational failure. No intelligent man would be surprised if such boys failed to reach their full possibilities.

Any statement such as these and any statement made in the Report concerning the salaries of engineers must therefore be interpreted in the light of the wide-spread definition of activity included in the term "engineering." There is and always will be a large, amount of engineering work that can be done by mediocre men, just as there are corresponding activities in all branches of commerce and industry. I have an idea that if the average pay of all men who are dignified by the title of "manager" was computed it would not cause a wild rush for jobs so dignified. But again, no one can view our present civilization with a prophetic eye and not be impressed with the vast amount of work in industry and commerce that demands the highest of engineering training and for which there is and will be adequate reward. For a number of years practically all of the members of the graduating class in engineering at Cornell University, numbering approximately 250, have been engaged before graduation at salaries ranging from $100 to $125 per month. This condition is not peculiar to Cornell, but is found at all the higher-grade institutions. This certainly is a good start for a young engineer (or even for a stenographer), and his future success lies entirely in his own hands and depends upon his own efforts. We shall known much more about the degree of success these young men attain when the inquiry now being made by the Society for the Promotion of Engineering Education is completed. In the meantime we have nothing definite upon the subject, though we have many *opinions* like those quoted in the foregoing and like the following from the Report:

> Salaries have risen, but less than the cost of living. It is more difficult to convey a general idea of salaries in the United States than in almost any other country. A beginner who wishes to obtain training and experience may accept $1000 a year, while the leading mechanical engineer on a railway may earn $40,000 to $50,000 a year. An ordinary engineer may earn anything from $5000 to $25,000. The real salaries of public officials are estimated to be at about 70 per cent of their prewar value.

There is another phase of this problem that should be noted, namely, the charge that the colleges are graduating too many men of inferior ability who cannot be developed into high-class engineers and who, filling lower levels in industry, produce the group of engineers referred to in Mr. Kelly's quotation. Again we shall know more about this matter when the S.P.E.E. completes its investigation, but there is good ground for suspicion that this may be true not only of technical schools, but of colleges in general. So long as colleges and universities continue to admit poorly prepared men and women, so long will they continue to send out some poor product. Yet the remedy is easy. A rigid enforcement of good academic standards, both at entrance and during college training, would greatly reduce the large numbers of students that now beset some institutions of learning. It is true that this would remove some reasons for boasting as to numbers of students, in fact it would be in the nature of a major operation for some institutions; but the results would be healthful in the long run and there would probably be fewer discontented professional men.

Sections IV and V of the Report are both short and contain little of startling interest. Section IV discusses insurance and provident institutions, and is interesting only in that it presents a sketch of what is being done in other lands regarding compulsory accident and sick benefits. Section V discusses very briefly certain efforts of engineers in a few other lands to form protective associations. On the whole the Report is well worth reading, for while it is necessarily sketchy and incomplete, as the authors state—"One of the results of the inquiry may be to improve the international and intellectual relations between the workers to whom it refers, and this may be said to constitute its main object."

DEXTER S. KIMBALL.[1]

Community Planning

THE need for better-designed, more beautifully executed communities is receiving increasing attention by engineers. This was shown at the conference in New York from April 20 through 25 which was attended by representatives of twenty foreign nations and three hundred from the United States, among whom there were a goodly number of engineers. It is fitting that this is so, for the design of a community, be it town, city, state, or an economic unit such as the Port of New York, which touches three states, is essentially an engineering task. A community that has an economic and social purpose to fulfil should be designed to execute that purpose economically with due regard for the safety and comfort of human life and the elevation of its ideals. The location of the industrial, commercial, residential, governmental, and social components of a community and the selection of the means and methods of transportation, communication, and power distribution between these components and in turn between the entire community and other communities, demand engineering vision. As is often the case in engineering design, the best achievable result must be a compromise between the desired ideal and things as they are, and the engineer is known to be expert in securing the best possible result at the lowest cost.

There are many phases of community design to which the mechanical engineer can make important contributions from his knowledge and experience. Some of these are the possibility of grouping industries so that by the joint use of power and heat the consumption of fuel may be reduced; the arrangement of residential districts for the best use of central heating, with consequent fuel economy; and the relative location of the residential and industrial districts so that power distribution costs will be minimized and service will not be impaired. The passenger-transportation facilities between homes and industries, the freight handling to and from industrial districts, and the provision of attractive residence districts with their beneficial effect on labor turnover, are of distinct importance to the mechanical engineer as the engineer of industry. The engineers of prominence who have made contributions to the art of community planning have been civil engineers who were brought into the work through their contact with the engineering problems of cities or transportation systems. Mechanical engineers should bring their knowledge, experience, and influence to the support of those already in the field that the work may be further advanced and the public made more fully aware of its importance and desirability. As community-planning problems are essentially local, such activity may best be effectuated through the local engineering societies and the Local Sections of The American Society of Mechanical Engineers. The dissemination of facts and the presentation of sound principles of community design afford an excellent opportunity for the coöperation of the engineers in a community for public service. Several local engineering groups have made notable contributions in this field, among them those in Detroit, Milwaukee, and Cincinnati, but there are still many communities which need engineering leadership for development along sound lines.

The outstanding influence which now affects every phase of community planning is motor traffic. It is estimated that the

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automobile market will not approach saturation until there is an average of one car for each family in the United States, or in the year 1940, thirty million cars. In 1920 over half the people of this county lived in cities of over 2500 inhabitants, and the percentage is increasing. The cities are thus facing a problem which cannot be solved by sweeping property changes which may require forty years for completion. Better use of existing facilities is imperative, and this may be accomplished in each community through the efforts of a strong civic body under engineering leadership.

Aeronautic Research to be Undertaken by American Engineering Council

THE American Engineering Council, through its Administrative Board, meeting at the Philadelphia Engineers' Club, May 8 and 9, voted to undertake what is described as the first exhaustive investigation of the aircraft situation yet made in this country. The various aspects of the problem, military and civil, will be thoroughly sifted by a special committee, which is expected to occupy one year in the task at an estimated cost of $50,000.

The Board's action followed the adoption of the report and recommendation of the Council's Committee on Aircraft, presented by Prof. Joseph W. Roe, head of the Department of Industrial Engineering in New York University and chairman of the Committee. The report warned that the present confused situation is a national menace both as to preparedness and as to commercial development. Great danger to investors exists, it was asserted, owing to the uncertain conditions which prevail.

The report recommended the following resolution in regard to the Air Mail which was unanimously adopted.

WHEREAS, The engineering profession of the United States is aware that the early establishment of a self-supporting civil industry in the field of aviation is of greatest importance; and,

WHEREAS, Determination of the sound economic field of commercial aviation, and development in this field, are greatly to the interest of the industry and business of the United States; and,

WHEREAS, The transportation of mails by aircraft offers an unusual opportunity to further both of these ends, as the Post Office Air Mail Service has admirably demonstrated; and,

WHEREAS, Preparedness in aviation is a vital element in national defense,

Be it Resolved, That the American Engineering Council, through its Administrative Board, bring its best influence to bear on Congress, the Post Office Department, and any other Government agencies involved, to give every possible consideration and encouragement to the transportation of mails by aircraft; and

That the Board pledges its influence in encouraging the widest utilization of the service so afforded.

In furtherance of the last paragraph of the preceding resolution, the Board passed the following second resolution:

WHEREAS, The Air Mail Service is rendering valuable service to the business and industry of the United States, and creating a reserve for national defense, and should receive the fullest support,

Be it Resolved, That the American Engineering Council recommend to the Engineering Societies of the United States and to their members that they:

(*a*) Encourage the work of the Air Mail and utilize its service wherever possible

(*b*) Bring the attention of members to the Service through their meetings, publications, etc.

(*c*) Spread information of the facilities offered, procedure in utilizing them, the time saved, postal rates in force, etc.

(*d*) Ask their Senators and Congressmen to further legislation for the development of the transportation of mail by aircraft, and

That the secretary's office have available for member societies and others, information as to the lectures, films, reports, etc. upon the Air Mail and civil and military aviation, which may be used in connection with the above.

Professor Roe further advised the Board that there is not available at this time any general study of the subject of civil aviation in the United States, and the Committee recommended to the Board that it further a comprehensive study and report on the present status and possibilities of commercial aviation.

In scientific and engineering progress in aviation (said the report), more particularly in heavier-than-air machines, the United States now leads the world. In the development of commercial aviation by private initiative, however, the United States is lagging far behind other countries. It suffers on one hand from the representations of enthusiastic promoters whose claims arouse suspicion; on the other hand from a general indifference due to ignorance of what has been actually accomplished, of the development, reliability, safety, etc. of commercial aviation in Europe and of the services which aircraft can render to modern business.

If commercial aviation is to be developed in this country it will probably be largely through private financial support rather than governmental aid, as in Europe. This support must come from investors and banks, who will not and cannot be interested in this field until the facts relating to the commercial possibilities, risks, etc. are available.

The Committee felt that a valuable service would be rendered the country and future business if some proper agency, which commands attention and confidence, will make a searching and impartial study and report of this field. If it is inherently unsound for investment, the public should know it. If, under proper conditions, certain services offer opportunity for investment, these services and the conditions necessary should be made clear. The study proposed should be conducted in a spirit of scientific research, with openmindedness and absolute adherence to fact. Only so can it aid the public and guide investment policy.

The study should be confined to the operation of aircraft, as distinct from manufacture. We suggest that it might cover:

a The status of commercial aviation in other countries, its extent, the varieties of service offered, the volume of business done, safety, regularity, financial status, extent and character of government aid, etc.

b A study of the geographical, traffic, and other conditions necessary for the successful transportation of mail, express, and passengers, and the conditions under which air routes must operate for reasonable promise of commercial profit.

c A study of the types of machines best adapted for different services.

d A study of the flying fields, hangars, and other facilities required which must or might be owned and operated privately.

e A study of air legislation such as licensing of pilots, rules of the air, etc.

f A study of the Government aid which can and should be given to commercial aviation in the form of beacons, landing lights, route markings, radio direction and meterological service, etc. analogous to the harbor improvements, lighthouses, coast services, etc. provided for water navigation.

g A study of the investment possibilities of such routes, of the capital required, financial risks, and the possible income and earnings.

h A study of the benefits likely to accrue in other fields from the establishment of such routes, as in banking through saving of interest on notes, exchange, etc.

i A study of the uses other than transportation, to which aircraft have been and may be put, such as aerial photography and surveying, forest patrol, protection against boll weevil, etc.

j Other phases which might develop during the course of the investigation.

The proposed study has two phases. The first is fact finding, bringing forth in effective manner statistics, facts, etc. now existing but not generally available; the second is constructive investigation developing new facts, methods of procedure, etc. not as yet formulated. The first group in the recommendation is of the former character. Most of the remaining fields combine the element of research, although for some of them there is accumulating experience which would come within the scope of a purely fact-finding survey. An impartial fact-finding survey, alone, would be of value, having two definite advantages: it limits the investigation to existing *facts* and thus avoids entering controversial fields, and defines the scope of the inquiry. The handicap, however, of such limitation is that the body of actual commercial operating experience is very limited, and the answers we most want lie beyond the mere collection and assembling of existing data. The solution of these further problems would throw much light on the potential field for aircraft operations.

The aircraft-manufacturing industry should not be asked to finance it, even if in its present precarious position it were able to do so. An investigation so financed would be subject to attack, however, just its findings. While the Army and Navy have many of the facts and have men qualified for the work, the country at large would look upon them as interested parties, and upon their findings as a brief for a client rather than a judicial decision.

Such an investigation is needed. The American Engineering Council is qualified to formulate the problem and develop the scope of the inquiry. If it could finance such a study there is a possibility of rendering in this field a service similar to that accomplished by the Waste Report and the 12-Hour-Shift Report.

A spirited discussion took place among the engineers over the question whether the preparedness factor should be stressed in the inquiry about to begin. Dr. H. E. Howe of Washington, treasurer of the Council, declared that the possibility of war should not be ignored, and that Japan and Germany were working closely together in developing chemical warfare. Prof. C. F. Scott of Yale expressed the view, shared by several other members of the Board, that the investigation should emphasize commercial aviation. The resolution as finally adopted provides for the consideration of both civil and military conditions.

The present Committee on Aircraft of the Council was dismissed, its work having been completed with the presentation of the report. The president of the Council, former Governor James Hartness of Vermont, will appoint a new committee to carry on this investigation under the direction of the Administrative Board of the Council.

Work will begin at once, the first task of the Committee being the provision of funds and the mapping out of procedure. The Committee will report at the next meeting of the Board to be held in Columbus, Ohio, in October.

American Forest Week

IT IS significant that of the seventy-nine associations which took part in the official program of American Forest Week, April 27 to May, 3, eleven of them were technical organizations or had technical elements. This shows that the efforts of those responsible for the organization and development of the Week four years ago to broaden the appeal from just that of forest-fire prevention to all the interests connected with forests and their problems, are meeting with success.

Wood, in its original form, or chemically or physically altered, enters into more than 2000 articles of daily use, in the preparation of which there is no suitable substitute.

The appeal of American Forest Week was therefore this year fortunately made on the basis not only of the tangible but also of the intangible benefits of our forests. The "tangibles," to the general public are the planting of trees and their care. The "intangibles" are, from the public's point of view, the economic use of forest products in the great manufacturing industries. Hence the bringing in, this year, in this popular movement of a proportion of the technical societies.

Furniture making; the manufacture of some fifty different large groups of tools, vehicles, machines, and articles that are preferably made of wood; the production of turpentine and rosin; certain chemical industries; a large group of industries that use pulp as their raw material for other purposes than papermaking—these intangibles to the public are very tangible to the technologist and the engineer.

From the point of view that the American Forest Week Committee had the vision to draw into its ranks a large group of persons representative of every phase of public interest and dependence upon the forests, and to broaden the purposes of its "Week," it may be said that a successful beginning in this year's event has been made.

There is still great work to be done. The public organizations now have it well in hand to make the people of this country "forest-minded." Our program—the program of the technical societies—is to take up the technical questions of conservation and waste utilization, for which problems the American Forest Week Committee recognizes an aptitude by broadening its purposes to include our scope.

The Hazards of Tetraethyl Gasoline

IT HAS BEEN known for a long time that in internal-combustion engines of the type used in motor cars and airplanes, increased output can be obtained by raising the compression ratio. It has also been known, however, that as the compression ratio in an engine is increased, the danger of preignition increases also, the compression pressure being high enough to raise the temperature of the mixture to a point where, given certain other conditions, such as poorly cooled spots in the combustion chamber, ignition of the mixture will take place before the spark occurs. If this happens the engine will lose in power output instead of gaining, and as this condition is manifested also by a clearly distinguishable sound, it has been referred to as "knocking."

During the War the Army Service in coöperation with the Dayton Research Laboratories, which latter have since become the General Motors Research Laboratories, have for the first time thoroughly investigated this phenomenon. It was found that preignition could be prevented by giving the fuel a special chemical structure, obtained, for example, by using mixtures of ordinary aviation gasoline with cyclohexane alone or in certain combinations. Another way to prevent such preignition and thus raise the permissible compression ratio with consequent increase of power output of the engine was by introducing into the fuel minute quantities of materials such as iodine. This research was vigorously continued after the War by the General Motors Research Laboratories under the direction of Chas. F. Kettering and Thos. Midgley, Jr., with the result that it was discovered that a chemical hitherto known practically only as a laboratory curiosity, namely, tetraethyl of lead, together with certain other materials, such as sodium, bromine, etc. produces the same or even better effects than iodine and permits the operation of the engine at compression temperatures which would be impossible with untreated gasoline. The amount of the lead-compound addition is said to be in the ratio of one-tenth of one per cent. It need hardly be said that this opens extensive opportunities for a basic redesign of automobile engines along lines of higher compression ratios, with consequent increase in power output and economy in fuel consumption.

Since tetraethyl of lead, as the material may be called for the sake of simplicity, has been placed on the market, however, vigorous objections have been made to its introduction, it having been claimed that its extensive use may prove to be a grave menace to public health. These objections received a dramatic confirmation in occurrences at the Bayway plant of the Standard Oil Co. of New Jersey, one of the owners of the tetraethyl-of-lead process, when several men died of an acute and previously comparatively unusual form of lead poisoning, while many others were seriously incapacitated.

There is no question that lead when introduced into the human body is apt to produce the most deleterious results, a fact which has been known for at least 2000 years. J. T. Arlidge in his well-known "Diseases of Occupation" states as follows:

> The poison finds its way gradually into the whole mass of the circulating blood and exerts its effect mainly on the nervous system, paralyzing nerve forces and with it muscular power. . . . . The functions of the stomach and bowels are deranged. . . . . Other organs frequently involved are the kidneys, the tissue of which becomes permanently damaged, whilst the sight is weakened or even lost.

One of the worse features of lead poisoning is that apparently the average human body has only little capacity for getting rid of any lead which has gained access into it. Because of this the effect of lead is cumulative, and even extremely minute amounts if absorbed frequently enough may in the end produce the most terrible effects. One of these effects called "lead encephalopathy," was discussed in a treatise on Lead Diseases by L. Tanquerel des Planches, published about 1835. It may appear in various forms such as tranquil or furious delirium, comatose state, convulsions or epilepsy, many of these having been observed in the victims at the Bayway plant of the Standard Oil Co. These and other considerations led several prominent men, such as Prof. Yandell Henderson of Yale University, Dr. Chas. L. Dana (in a paper before the American Neurological Association, Washington, May 6, 1925), and Professor Zangger of Zurich, Switzerland, to express the opinion that extensive use of this material may cause great harm to the health of the people. The Bureau of Mines made a quick test on tetraethyl of lead, using small animals as objects of experimentation, but while it is known that cats and horses succumb to lead poisoning if exposed for a sufficiently long period, practically nothing is known as to the susceptibility of animals in general to this form of poisoning. Furthermore, even in the case of humans exposed to lead poisoning, individual characteristics prove of immense importance, with the result that under the same conditions some men may work for scores of years without feeling any harmful effects, while others are seriously affected within a period of a few weeks. Any test to be convincing should be carried out only on animals of a type known to be susceptible to lead poisoning in approximately the same degree as humans, and should be carried out on a number of animals sufficiently large to eliminate the influence of individual characteristics and for periods sufficiently long to insure the reliability of the results obtained.

It is fortunate that this matter is apparently in the right hands at the present moment, Dr. Cummings of the U. S. Public Health Service having called a conference to determine the proper methods of handling the situation. Unquestionably suitable ways of experimentation will be determined at this conference and the problem will receive the attention which it deserves.

Engineers more than any other class will appreciate the splendid achievement represented in the work of Messrs. Kettering and Midgley, who were the first to gain a clear insight into the phenomenon of "pinking," and who, in the course of their investigations, developed some very remarkable experimental apparatus and

methods. The discovery of a material practically unknown until then and the development of this material from a laboratory curiosity into a possibly important industrial product is likewise an outstanding achievement of American engineering. If the objections to its use are merely such as are apt to be set against new and radical development, the sooner they are dispelled by the competent authority of the U. S. Public Health Service the better it will be; but if the use of tetraethyl of lead does involve appreciable danger to life, enough is known of the high moral standing of its inventors and the companies manufacturing it to warrant the statement that they themselves would be the very last to try to market it under such conditions.

Engineers Elected to Academy of Sciences

DURING its annual meeting at Washington, D. C., which terminated on April 29, the National Academy of Sciences elected twelve life members and six foreign associates, among whom are several engineers.

Elmer A. Sperry, Mem. A.S.M.E., was elected to life membership, as was also Ralph Modjeski, a member of the American Society of Civil Engineers. Sir Charles A. Parsons, Hon. Mem. A.S.M.E., and Eugene Schneider, whose father, Henri Schneider, was also an honorary member of the Society, were made foreign associates of the Academy.

The National Academy of Sciences was organized in 1863, with the approval of President Lincoln, and its Act of Incorporation says that it shall, "whenever called upon by any Department of the Government, investigate, examine, experiment, and report upon any subject of science or art." It is stipulated that the Government shall pay the actual expense of such investigations, but that the Academy "shall receive no compensation whatever for any services to the Government of the United States." Members must be American citizens. The membership was originally limited to fifty, but that limitation was removed three years later. Nevertheless the membership remains small, as only a few noted scientists are elected each year. There are at present about two hundred and fifty members and forty-four foreign associates. The outbreak of the war in 1914 led to the organization of the National Research Council by the National Academy, and this war service proved so valuable that President Wilson issued an order at the close of the war, requesting the National Academy of Sciences to continue the National Research Council. The Research Council, therefore, was reorganized on a permanent peace basis as an agent of the Academy, and shares the headquarters of the Academy at Washington. The Academy has Sections on Mathematics, Astronomy, Physics, Engineering, Chemistry, Geology and Paleontology, Botany, Zoölogy and Animal Morphology, Physiology and Pathology, Anthropology, and Psychology. Of the engineers elected this year, Elmer A. Sperry is noted especially for his gyro-compass, airplane and ship stabilizers, and high-intensity searchlight. He holds over four hundred patents covering these and his other inventions, which include a compound internal-combustion engine, fire-control apparatus, an electric chain mining machine, detinning and electrochemical processes, and machinery for making fuse wires. In 1883 he erected what was then the highest electric beacon in the world (about 350 ft.) on Lake Michigan.

Sir Charles A. Parsons, of London, England, is the inventor of the Parsons compound steam turbine, which was introduced about 1884, and also of a geared turbine which appeared about 1910. He is chairman of C. A. Parsons & Co., of Newcastle-on-Tyne, and of a number of other large companies in England.

Eugene Schneider is the head of the immense engineering and steel works at Le Creusot, France, which came to the fore so remarkably during the war, and is the inventor of the famous French 75-mm. gun.

Ralph Modjeski, civil engineer of Chicago, is a well-known builder of bridges. He was consulting engineer for the city of Chicago and the Sanitary District on their bascule bridges; designed and built the Government bridge at Rock Island, Ill.; designed and built the Columbia and Willamette River bridges for the Portland & Seattle Ry.; was chief engineer on the McKinley Bridge at St. Louis and on other large bridges, and is now chief engineer on the Delaware River Bridge at Philadelphia.

A.S.T.M. Establishes Medal for Research

FOR SEVERAL YEARS the American Society for Testing Materials has been giving considerable thought to means for stimulating research in engineering materials. As a result, it is proposed to establish an annual medal and an annual lecture, both of which, it is believed, will help materially in broadening knowledge of the properties of materials.

It is proposed that the medal shall take the name of the first president of the society, Charles B. Dudley, whose most distinguished service was as chemist of the Pennsylvania Railroad. Dr. Dudley, who served as president of the A.S.T.M. from 1902 to 1909, was a research worker and investigator of note, so that it is most fitting that this medal is to bear his name.

It is recommended that the Dudley Medal shall be awarded to the author or authors of a paper of outstanding merit presented before the society and constituting an original contribution on research in materials, with the understanding that if no paper in any given year seems to merit this distinction, the award will not be made; and that the Executive Commitee shall appoint annually a Committee on Award of the Medal to consist of three members.

Similarly, it is proposed that the lecture shall commemorate the name of the society's first secretary, Edgar Marburg. Serving as secretary from 1902 until his death in 1918, Dr. Marburg put the work of the society on a very firm foundation and, through his development of the technical programs year by year, brought wide recognition to the society as a forum for the discussion of properties and tests of engineering materials. His valued services in this direction and his own worthy contributions to the knowledge of materials are fittingly recognized in the proposal to identify his name with this lecture.

It is recommended that the lecture shall afford an opportunity for the society to be addressed by leaders in the field of engineering materials who will present outstanding developments in the promotion of knowledge of engineering materials, and shall carry with it an honorarium of $200 and a suitably engraved certificate. It is proposed that the Executive Committee appoint annually a committee of three members to select the lecturer, and that in this selection consideration be given to the plan of securing foreign lecturers at suitable intervals and as the funds permit.

These proposals, made by the society's committee on correlation of research, will be recommended to the society by the Executive Committee at the annual meeting in Atlantic City in June. The establishment of the medal and lecture will require about $6000, which it is proposed to secure through contributions from members of the A.S.T.M.

Francis Hodgkinson Honored by Franklin Institute

IN RECOGNITION of his achievements in the field of steam-turbine design and construction during the past thirty-one years, Francis Hodgkinson, chief engineer of the South Philadelphia Works of the Westinghouse Electric & Manufacturing Company, was recently awarded the Elliott Cresson Gold Medal. This medal, one of the highest honors conferred by The Franklin Institute, is given for "discovery or original research, adding to the sum of human knowledge, irrespective of commercial value; leading and practical utilizations of discovery; and invention, methods, or products embodying substantial elements of leadership in their respective classes, or unusual skill or perfection in workmanship."

Mr. Hodgkinson, who was born in England, received his primary education at the Royal Naval School at New Cross, London, finishing in 1882, and then began his apprenticeship, serving for the succeeding three years. In 1885 he joined C. A. Parsons and Company, the world-famous turbine-engine company, remaining with them until 1890, during which time he was engaged in steam-turbine work, first as machinist, later as draftsman, and finally as superintendent of field work.

Going abroad in 1890, Mr. Hodgkinson joined the Chilean Navy and served during the civil war of that time, rising from the rank of third engineer to that of second engineer of what was then a modern destroyer. Subsequently, he left the navy and became engineer for the telephone and electric light company of Lima, Peru, working in this capacity until 1892.

From 1892 until 1894 Mr. Hodgkinson was connected with a mining company operating properties at Casapalca, Peru, where he was in charge of maintenance. At the end of that time he returned to England, rejoining C. A. Parsons and Company as general foreman of erecting and fitting shops. The following year the Westinghouse Machine Company, the original name of the turbine-building section of the Westinghouse Electric & Manufacturing Company, secured the American rights for the manufacture of the Parsons-type turbines. In 1896 Mr. Hodgkinson, upon the recommendation of the Honorable C. A. Parsons, became connected with the Westinghouse Machine Company, taking charge of the engineering and designing of the Parsons-type turbines. This position he still retains and, in addition, since 1916 has been acting as chief engineer in charge of all the engineering departments of the South Philadelphia Works. Practically all the commercial steam turbines built by this company have been largely originated by him and designed under his direction.

Columbia University to Administer Patents

PATENTS arising from discoveries made in the laboratories of Columbia University may now be administered by the university under a recent ruling of its Board of Trustees. A Board of University Patents has been established and given authority to accept the assignment of patents, patent applications, royalties, licenses, etc. produced by members of the teaching staff or by use of the laboratories of the university or otherwise.

The central purpose of this action is by the coöperation of the university to protect the discoverer or inventor of a patentable article or process, to insure service to the public under best possible conditions, and to enable the university to share in the benefits of a patent so that funds at its disposal for the promotion of research may be increased.

Engineering Foundation Completes First Ten Years

MARKING the beginning of its second decennium, Engineering Foundation tendered a subscription dinner to its founder, Ambrose Swasey, at the Union League Club, May 14. Present and former members of the Foundation, the Trustees of the United Engineering Society, and the officers and directors of the American Society of Civil Engineers, the American Institute of Mining and Metallurgical Engineers, The American Society of Mechanical Engineers, and the American Institute of Electrical Engineers, numbering in all forty-five persons, participated in the event.

Following the dinner Vice-Chairman Edward Dean Adams, who presided in the absence of Chairman Stillwell, in behalf of the societies represented in the Foundation, presented Mr. Swasey with a testimonial expressing their gratitude for his vision and his gifts for the furtherance of research and the good of mankind. On behalf of the Foundation Board, Charles F. Rand, who has been Chairman for five years, was presented a resolution in appreciation of his services.

After the business meeting J. V. W. Reynders, of the A.I.M.E., voiced the esteem in which Mr. Swasey is held and the appreciation of the Founder Societies for his making possible the establishment of the Foundation. Dr. Frank B. Jewett, a former Vice-Chairman of the Foundation, then gave an address on Permalloy Cables for Submarine Telegraph, illustrated by a motion picture of the landing at the Azores of the first cable in which this new alloy is used.

The tenth annual report of Engineering Foundation reviews briefly the work accomplished during the past ten years and gives a statement of its activities during the year ended February 11, 1925. The following classification of expenditures shows how its income has been used:

Aid in establishing National Research Council	$10,500
Support of Division of Engineering, National Research Council	26,000
Research projects	67,800
Promotion of research	26,000
Administrative expenditures	28,100
Total expenditures in 10 years	$158,400

An unexpended balance January 1, 1925, of $39,800 makes the total income for the ten years $198,200.

A bequest to United Engineering Society in the will of Henry R. Towne, Past-President and Honorary Member of The American Society of Mechanical Engineers, who died October 15, 1924, provides for establishing the Henry R. Towne Engineering Fund of $50,000, the income of which is to be devoted to the general purposes of Engineering Foundation.

One item of research which has engaged the attention of the Foundation for a number of years is that on the fatigue of metals, which is being carried on at the Engineering Experiment Station of the University of Illinois, under the general supervision of a committee of the Division of Engineering and Industrial Research of the National Research Council. Prof. H. F. Moore, who is in charge of this work, gives the following summary of progress in the Foundation's tenth annual report:

> A well-equipped laboratory has been developed and skilled test workers have been trained so that results already achieved can be supplemented. A promising field for exploration is the fatigue strength of steel castings. Steel castings are coming into more and more common use for machine parts, and there are practically no data available to determine whether they possess as high fatigue strength as parts made of rolled or forged steel. Another inquiry recently initiated, which has large possibilities for useful results, is a search for methods of detecting fatigue failure in its early stages before the minute cracks which mark its beginning have spread far enough to cause the destruction of a machine or structural part.
>
> Existence of fatigue limits has been established and values have been determined for the common grades of wrought ferrous metals, including many alloy steels. The interesting fact has been brought out that these limiting stresses for wrought ferrous metals bear a definite relation to the ordinary tensile strength of the steel and to its Brinell hardness. Endurance limits have been determined for steel under repeated flexural stress, for steel under repeated shearing stress, and for steel under cycles of alternating tension and compression.
>
> At the end of 1924, the fact outstanding from the tests made on brasses and bronzes is that in comparison with ordinary tensile strength the fatigue strength is relatively much lower than that of wrought ferrous metals. Endurance limits for non-ferrous metals, if any exist, are not nearly so well marked as the endurance limits for wrought ferrous metals.

A.S.M.E. Spring Meeting at Milwaukee Breaks Records for Attendance

AS THESE PAGES go to press word comes from Milwaukee that at the close of the second day of the Meeting the registration totaled 930, which is the largest registration that has ever been recorded at any A.S.M.E. Spring Meeting. The sessions held thus far—Machine-Shop Practice, Hydroelectric, Forest Products, and Pulverized Coal—have been unusually well attended and the discussion valuable and spirited. The public hearing on the care of power boilers held on Monday afternoon, May 18, was of interest to many and in general the comments at the session were favorable. The reception, which was held in the Hotel Pfister on Monday evening, was most thoroughly enjoyed, those in attendance being welcomed by the Mayor of Milwaukee. Many took advantage of the opportunities afforded by the excursions to visit some of the nationally known industrial establishments of the city.

The Council of the Society has authorized a change in the name of the Forest Products Division: it will henceforth be known as the Wood Industries Division. San Francisco has been selected as the city in which to hold the 1926 Spring Meeting and Richmond, Va., for the October Regional Meeting of that year.

The Power Show which is being held in Milwaukee during the same week as the Spring Meeting, is proving an excellent attraction.

A general account of the meeting will appear in the July issue of MECHANICAL ENGINEERING.

Industrial Psychology

THE Journal of Industrial Psychology will issue its first number on January 1, 1926. It will be issued bi-monthly and the subject matter will include treatments of the problems of selection of employees, the motivation of industrial morale, and the elimination of fatigue. Dr. Donald A. Laird of the Psychological Laboratory of Colgate University is the editor. He will be assisted by a board of coöperating editors selected from industrial and scientific fields.

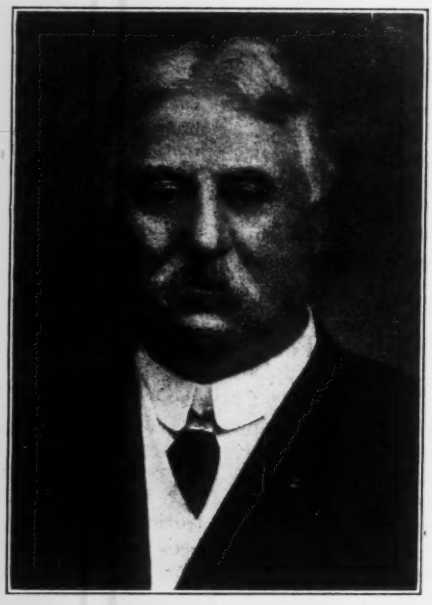
Major William H. Wiley Dies

Treasurer of The American Society of Mechanical Engineers for the Past Forty-One Years Passes Away at the Age of Eighty-Three

WILLIAM H. WILEY, Treasurer of The American Society of Mechanical Engineers since 1884, died in his eighty-third year at his home in East Orange, N. J., on Saturday, May 2, 1925, after an illness which started about the middle of January. He had been identified with the publishing world since 1876, in which year he joined his father, John Wiley, and his older brother, Charles, doing business under the firm name of John Wiley & Son.

Major Wiley was born in New York City on July 10, 1842, where he spent the first dozen years or so of his life. He was fond of telling how as a small boy, full of mischief, he was often packed off to Springfield, Mass., to receive strong moral discipline from his maternal grandfather, the Reverend Doctor Samuel Osgood, a famous Congregational preacher and a great abolitionist.

In the early fifties the family moved to East Orange, N. J. Major Wiley was attending the College of the City of New York when the Civil War broke out. He immediately joined the Independent Corps of the New York Volunteer Light Infantry as first lieutenant, an organization later known as the Seventh Regiment. Many years afterward he was elected president of the Seventh Regiment War Veterans.



WILLIAM H. WILEY

In the course of the War Major Wiley rose to the command of two companies of artillery, seeing most of his service in South Carolina where he served with distinction in a number of severe engagements. He was given special mention by General Gillmore "for efficient and able services during the siege and bombardment of Fort Sumter and Charleston, S. C." In 1864 he was retired as brevet major "for gallant and meritorious services."

After the War Major Wiley resumed his studies and in 1866 received the degree of Civil Engineer from Rensselaer Polytechnic Institute, following this course with a year of special work at the Columbia School of Mines. For the next nine years he practiced engineering in the East and Midwest, dropping this practice to enter the publishing business in 1876. The Wiley publications at that time included books of many sorts, but Major Wiley soon saw a great opportunity to develop the business into one of technical and scientific books. From his service in the War and as an engineer he noticed the great dearth of good American scientific books. His friendship with officers of both the Army and Navy made it natural that he should be selected both by West Point and Annapolis as the publisher of their books.

Major Wiley's wide and varied experience in the engineering field brought him into direct contact with the most prominent engineers of the country and gave him a first-hand knowledge of their needs. This knowledge enabled him to make many suggestions which led to the writing of needed engineering books by well-qualified authors. Thus the business became definitely one of a scientific nature.

Under his able direction, which continued until the last year of his life, the Wiley lists grew to include books in all the branches of engineering, agriculture, chemistry, geology, mathematics, and the biological sciences. In the last few years of his life, realizing the close link between engineering and business, he encouraged his associates to develop a group of books in the latter field.

Despite his great activity in the publishing field, Major Wiley found time for many other activities. For three years he was a member of the township committee of East Orange, serving for one year as its chairman. He served three terms in Congress as a representative from New Jersey. He was a commissioner from New Jersey to the St. Louis Exposition in 1904 and was president of the International Jury of the Brussels Exposition in 1907, being decorated with the Order of Leopold by the King of the Belgians. He was a member of the executive committee and during the World War was chairman of the National Preparedness Committee of The American Society of Mechanical Engineers. He was a member of the American Society of Civil Engineers, the American Society of Mining and Metallurgical Engineers, American Institute of Electrical Engineers, and The American Society of Mechanical Engineers. He belonged also to the American Association for the Advancement of Science. Major Wiley was a member of many clubs, including the University and Engineers' Clubs of New York, the Cosmos Club of Washington, and many others.

Major Wiley was one of the eighty men present at the meeting on March 15, 1880, at which The American Society of Mechanical Engineers was founded. Four years later he became the treasurer of the Society, the third to hold that office, which was his to the time of his death, a record of forty-one years of unparalleled service to the organization.

In a word of personal appreciation, Dr. D. S. Jacobus, past-president of The American Society of Mechanical Engineers, writes as follows:

"He was a man of close and strong friendships. A home lover. To him it mattered little what others thought. There was always that chivalry and consideration which bespeaks a true gentleman. Above all he had the quality that convinced one of absolute truth and frankness which, with his kindly nature, endeared him to all. In the early days of the Society he was ever ready to lend a helping hand, and he smoothed out and remedied many an unpleasant situation. His earnest efforts and foresight contributed much to the upbuilding of our Society and he stands out as a typical example of the pioneers who have given freely of their best efforts to make it what it is today."

There is a story told of Major Wiley by Dr. George F. Kunz, which says that at one time the late "Boss" Tweed asked Major Wiley to become one of his lieutenants at a salary of $10,000 a year, but Major Wiley informed him he was in the book business and intended to remain in it. Tweed then told him that he was perfectly willing to have him remain in the book business, but wanted him as one of his lieutenants so that he could call on him occasionally when he needed him, and that the salary would still be $10,000 a year; but Major Wiley refused the offer definitely. It showed that Tweed recognized in him a man who was not only attractive and lovable, but also a good mixer and a man who might easily be able to influence legislators and others whom he himself could not hope to reach.

Mention should be made of the work that Major Wiley did in the early days of reporting the meetings of The American Society of Mechanical Engineers in London *Engineering*. These reports were models for accounts of the sort and did much to bring the work of the Society before the world at large. As an example of these reports may be cited the descriptive accounts which he wrote of the A.S.M.E. Meeting held in Washington, D. C., May 9 to 12, 1899, and which were published in six large installments in issues of *Engineering* running from July 7 to August 11 of that year.

Major Wiley was a man of large heart and keen, broad mind. His charity was without regard to position, creed, or color. For years after the Civil War he found employment for men who had served under him. He was a man loved and admired by all who knew him.

Book Reviews and Library Notes

THE Library is a coöperative activity of the A.S.C.E., the A.I.M.E., the A.S.M.E. and the A.I.E.E. It is administered by the United Engineering Society as a public reference library of engineering and the allied sciences. It contains 150,000 volumes and pamphlets and receives currently most of the important periodicals in its field. It is housed in the Engineering Societies Building, 29 West 39th St., New York, N. Y. In order to place its resources at the disposal of those unable to visit it in person, the Library is prepared to furnish lists of references to engineering subjects, copies of translations of articles, and similar assistance. Charges sufficient to cover the cost of this work are made.

The Library maintains a collection of modern technical books which may be rented by members residing in North America. A rental of five cents a day, plus transportation, is charged. In asking for information, letters should be made as definite as possible, so that the investigator may understand clearly what is desired.

Books Received in the Library

ANNUAIRE POUR L'AN 1925. By France. Bureau des Longitudes. Gauthier-Villars et Cie., Paris, 1925. Paper, 4 × 6 in., 884 pp., tables, 6 francs 50.

In addition to the customary data respecting the calendar, geodesy, astronomy, weights and measures, geography, population, etc. this almanac for 1925 contains a paper on the influence of the rotation of the earth on the physiognomy of the tides, by E. Fichot, and one on the applications of the three-electrode vacuum tube to astronomy, by G. Ferrié.

ARBEITSKUNDE. Edited by Johannes Riedel. B. G. Teubner, Leipzig and Berlin, 1925. Cloth, 6 × 9 in., 364 pp., 15 gold marks.

This volume is a discussion by a number of experts of various phases of the labor problem in both its broad aspects and its details. Section one deals with the history of work and the life of labor, the hygienic consequences of modern economic developments, economic psychology, labor politics and scientific management. Section two describes the scientific principles, anatomical, physiological, which should be known by the student of work. Section three discusses special problems, such as hygiene of workshops, tools, working hours, training, trade schools, management, methods of research, etc. The book will interest students of management and investigators of labor conditions.

AUTOMOBILE ENGINES. By Arthur W. Judge. Chapman & Hall, London, 1925. (Motor Manuals, v. 1.) Cloth, 5 × 8 in., 189 pp., illus., diagrams, 4 s.

The first of a series of manuals which will present the elementary aspects of automobiles in a simple, non-technical manner. Intended especially for owners and users who wish sound information on the theoretical and practical sides of the subject. This volume is confined to the engine. It includes the theoretical principles involved, an account of the different types in use and of their components, and sections on such supplementary topics as testing, lubrication, cooling, and maintenance.

JOHN A. BRASHEAR. . . . AUTOBIOGRAPHY. Edited by W. Lucien Scaife. The American Society of Mechanical Engineers, New York, 1924. Cloth, 6 × 9 in., 262 pp., illus., portraits, cloth, $4.25.

The autobiography of a man without formal education who started life as a journeyman machinist and rose to eminence through his great mechanical skill, as the foremost maker of astronomical lenses of his time. Dr. Brashear tells the story of his early days, of the growth of his interest in astronomy and of his struggles to build a telescope, in interesting, graphic language. His later career as a manufacturer, as an educator and as a man of public affairs is covered adequately to the close of his account in 1917. The editor has added a number of letters that continue the story to the time of his death in 1920.

CHEMISTRY IN THE SERVICE OF MAN. By Alexander Findlay. Third edition. Longmans, Green & Co., London and New York, 1925. Cloth, 5 × 8 in., 300 pp., illus., portraits, $2.

Intended to give readers without chemical knowledge, a readable, intelligible account of some of the more important general principles and theories of chemical science and of their applications. The author hopes that it may give the general reader some idea of his indebtedness to the chemist and also stimulate the interest of young students of chemistry. Three new chapters have been added to this edition, and various portions revised and rewritten.

THE COLUMBIA UNVEILED. By M. J. Lorraine. Times-Mirror Press, Los Angeles, 1924. Cloth, 5 × 8 in., 446 pp., illus., map, portraits, $3.50.

In June, 1921, Mr. Lorraine, a member of the American Society of Civil Engineers, built a rowboat at the source of the Columbia River and began a successful solitary journey to its mouth, some twelve hundred miles distant. This is the only authenticated continuous voyage to date, and Mr. Lorraine is apparently the second man to have covered the entire distance by boat.

His narrative is a plain, simple account of his experiences day by day, well illustrated by photographs along his route and accompanied by brief observations on the topography and geology of the country. Even though no attempt is made to lay stress on the dangers encountered, the story is an entertaining one, and the journey a remarkable feat for a man of sixty-seven.

CONSOLIDATION OF RAILROADS. By Walter M. W. Splawn. Macmillan Co., New York, 1925. Cloth, 5 × 8 in., 290 pp., $2.50.

The author of this work, as a member of the Texas Railroad Commission, has given much time to the study of the problems of consolidation and to the consolidations provided by the Transportation Act of 1920. The results of his study are recorded in this book, which shows the genesis of the idea of consolidation, tells what Congress expected to accomplish, and gives the opinion of the author as to the extent to which its expectations seem likely to be fulfilled, in the light of the record and arguments submitted to the Interstate Commerce Commission.

THOMAS ALVA EDISON. By Francis Arthur Jones. Thomas Y. Crowell Co., New York, 1924. Cloth, 5 × 8 in., 399 pp., illus., portraits, $3.

A popular anecdotal account of Edison, which deals primarily with the man rather than with his inventions, although the latter are not neglected. This edition has been revised, reset, and continued down to the present year.

EINSTEIN'S THEORY OF RELATIVITY. By Max Born. E. P. Dutton & Co., New York, 1922. Cloth, 6 × 9 in., 293 pp., diagrams, portraits, $5.

An elaboration of lectures given to an audience not conversant with mathematics. The book is a serious, well-executed introduction to the subject, in which the origin and development of this theory from the physical theories of Einstein's predecessors are traced. The author has avoided mathematics as far as possible and has used none but very elementary formulas.

EXAMPLES IN BATTERY ENGINEERING. By F. E. Austin. The author, Hanover, N. H., 1917. Cloth, 5 × 8 in., 90 pp., illus., diagrams, $1.25.

This textbook considers those matters that are important in the efficient operation of electric cells and batteries of all types. The fundamental principles are set forth and their application illustrated by examples which are solved in detail.

FIELD ENGINEERING. By William H. Searles. 19th edition, reviewed and enlarged by H. C. Ives. John Wiley & Sons, New York, 1925. Fabrikoid, 4 × 7 in., 2 vol. in one, diagrams, tables, $4.

This pocket-book, which has been in use for forty-five years

and has gone through many editions, is too well known to need description. The nineteenth edition has been so extensively revised as to necessitate a resetting of the entire text. Certain portions have been rearranged, some sections have been omitted and much new material has been added. The book still remains, however, small enough to be carried conveniently in the pocket, even when the two volumes are bound together.

LES FOYERS DE CHAUDIÈRES. By André Turin. Second edition. Dunod, Paris, 1925. (La Chaufferie Moderne.) Paper, 7 × 10 in., 467 pp., illus., diagrams, 63 francs.

An extensive treatise on boiler furnaces and their accessories. Covers hand-fired furnaces, furnaces for special fuels, mechanical stokers, furnaces for oil and gas, briquetting, etc. This edition has been widely revised in the light of changed conditions in France since the World War and is now representative of present practice in that land.

HENDRICKS COMMERCIAL REGISTER, 1925. S. E. Hendricks Co., New York, 1925. Cloth, 8 × 11 in., 2522 pp., $12.

The 1925 edition of this popular directory follows the plan of preceding ones but has been revised throughout. In the main section over seventeen thousand articles used in the engineering trades are listed, with the names of makers and dealers. Other sections contain an alphabetical directory of firms and a list of trade names. The book is carefully planned to meet the needs of purchasing agents.

HENLEY'S TWENTIETH CENTURY FORMULAS, RECIPES AND PROCESSES. Edited by Gardner D. Hiscox. Norman W. Henley Publishing Co., New York, 1924. Cloth, 6 × 9 in., 807 pp., $4.

It having become necessary to reprint this popular work, the occasion has been seized to include a new chapter on general methods for compounding recipes. The book contains several thousand recipes and processes for substances used in the workshop and the household.

INTERNAL-COMBUSTION ENGINES. By Wallace L. Lind. Second edition, revised. U. S. Naval Institute, Annapolis, Md., 1925. Cloth, 6 × 9 in., 266 pp., illus., diagrams, $3.

The author has aimed to provide a practical, up-to-date text suited to the comprehension of the average student and planned for those who will operate such engines rather than design them. The new edition has been thoroughly revised by the elimination of obsolete matter and the addition of new matter pertaining to aircraft, Diesel, and the Navy Type V-engines. The work is apparently intended primarily for use in the United States Naval Academy.

LABORATORY MANUAL OF EXPERIMENTS IN PHYSICS. By Leonard Rose Ingersoll. McGraw-Hill Book Co., New York, 1925. Cloth, 6 × 8 in., 220 pp., diagrams, tables, $2.

This manual, prepared primarily for students at the University of Wisconsin, contains a graduated series of experiments on mechanics, heat, electricity, magnetism, wave motion, sound, and light, which vary from those suited to students without previous training to experiments suited to sophomores.

LEHRBUCH DER METALLOGRAPHIE. By Gustav Tammann. Third edition. Leopold Vöss, Leipzig, 1923. Paper, 6 × 9 in., 450 pp., illus., diagrams, tables, $3.40.

In this book, the Director of the Göttingen Institute for Physical Chemistry offers a text on the theoretical aspects of the science of metals, adapted for the use of mature students. The basis of the volume is in general his own extensive researches and the arrangement of the material, which differs from the usual one, is intended to present the information so that it will emphasize the points of theoretical importance. This edition has been enlarged by the inclusion of the results of recent investigations in metallography.

LES NOTIONS FONDAMENTALES D'ELEMENT CHIMIQUE ET D'ATOME. By Georges Urbain. Gauthier-Villars, et Cie, Paris, 1925. (Science et Civilisation.) Paper, 6 × 8 in., 172 pp., diagrams, tables, 10 francs.

This book is an attempt to give a general account of our present knowledge of the constitution of matter. The idea of the element, radioactive elements, the constituents of the atom and its constitution, the relation between the structure of the atom and its properties, and isotopes are the subjects successfully treated.

The work, like the others in this series, is intended for readers with general interest in the subject rather than for specialists.

LIGHTING FIXTURES AND LIGHTING EFFECTS. By M. Luckiesh. McGraw-Hill Book Co., New York, 1925. Cloth, 6 × 9 in., 330 pp., illus., $4.

The author discusses lighting from an esthetic, rather than a commercial or industrial point of view. The effects of color and of painting, the design of fixtures, methods of lighting and the decorative uses of light are given attention. Throughout the book, the author shows how scientific control of light may be secured without loss of decorative quality.

A LONG LIFE'S WORK, AN AUTOBIOGRAPHY. By Sir Archibald Geikie. Macmillan & Co., London, 1924. Cloth, 6 × 9 in., 426 pp., portraits, $7.

Sir Archibald Geikie was born in Edinburgh in 1835 and educated in that city. Early attracted to geology, he entered Geological Survey in 1855 and began his long connection, which did not end until 1903, when he retired as Director-General. The next ten years were occupied by duties, first as Secretary, then as President, of the Royal Society. His autobiography is an interesting, modest account of his personal life and scientific career, containing much to interest the geologist.

METALLURGY OF ALUMINIUM AND ALUMINIUM ALLOYS. By Robert J. Anderson. Henry Carey Baird & Co., New York, 1925. Cloth, 6 × 9 in., 913 pp., illus., diagrams, tables, $10.

Since the last revision of Richards' classic work in 1896, no book has been published dealing with the metallurgy of aluminium and its alloys, and there is evident need for a modern work on this important metal. Mr. Anderson treats the subject from the point of view of the metallurgical engineer concerned with the production of the metal and its alloys in raw or manufactured form, and endeavors to avoid both the academic and the over-practical viewpoint. The book is comprehensive in its scope, covering the production of aluminium and aluminium alloys, their physical and chemical properties, methods of working, heat treatment, uses, etc. Good bibliographies are given.

MOLYBDENUM, CERIUM AND RELATED ALLOY STEELS. By H. W. Gillett and E. L. Mack. Chemical Catalog Co., New York, 1925. (American Chemical Society. Monograph series.) Cloth, 6 × 9 in., 299 pp., illus., diagrams, tables, $4.

As the United States has large reserve supplies of molybdenum, while the domestic supply of the other metals that are important in making alloy steels is small, the desirability of developing the molybdenum steels is of obvious importance to American steel makers and users. This book reviews and extends our knowledge of the possibilities and limitations of molybdenum as an alloying element, and presents a record of new investigations of its effect on endurance and impact properties and on the properties of "transverse" specimens. An extensive bibliography is given.

ORIGIN OF CONTINENTS AND OCEANS. By Alfred Wegener. Translated from the third German edition by J. G. A. Skerl. E. P. Dutton & Co., New York, 1924. Cloth, 6 × 9 in., 212 pp., illus., maps, $4.50.

A presentation in English of the author's "displacement theory" of the origin of continents and oceans. Professor Wegener's hypothesis is that the continents are of lighter material and float like icebergs on a heavier plastic, that the poles are not fixed relative to the plastic, but have occupied widely different positions, and that the land masses are moving away from the poles and westward. This movement has caused the separation of the continents. The book is devoted to the detailed establishment of this thesis.

PERSONNEL MANAGEMENT ON THE RAILROADS. By Policyholders' Service Bureau, Metropolitan Life Insurance Co. Simmons-Boardman Pub. Co., New York, 1925. Cloth, 5 × 8 in., 227 pp., illus, $3.

This study is concerned with the important problem of railroad administration: that of reconciling the antagonisms between railroad labor and management. The material of the report was gathered by reviewing the information in print, by surveying the methods of employee management used by eight railroads in different sections of the country, and by sustained contact with the

experiments in progress on the Baltimore and Ohio and the Rock Island railroads. The report attempts to indicate the outstanding elements of the personnel problem and to suggest measures which should improve conditions.

PRACTICAL ACCOUNTING AND COST KEEPING FOR CONTRACTORS. By Frank R. Walker. Frank R. Walker Company, Chicago, 1924. Cloth, 9 × 12 in., 169 pp., illus., $2.50.

The author has aimed to describe methods that will give all the information required with the minimum amount of clerical labor. Methods are given which are simple enough to be used by small contractors and yet can also be adapted to those engaged in business on a large scale.

QUATRE CONFÉRENCES SUR LA THÉORIE DE LA RELATIVITÉ. By Albert Einstein. Gauthier-Villars et Cie., Paris, 1925. Paper, 6 × 9 in., 97 pp., 12 francs.

A French translation from the German edition of Einstein's lectures at Princeton in 1921, which have also appeared in English.

RADIOACTIVITY AND THE SURFACE HISTORY OF THE EARTH; being the Halley Lecture, May, 1924. By John Joly. Oxford University Press, London, 1924. Paper, 6 × 9 in., 40 pp., diagrams, maps, $1.35.

In this interesting lecture, Dr. Joly presents clearly and concisely the modern view of the surface activities of the earth and discusses the nature and source of the energy that causes these activities.

RAILWAY ELECTRIFICATION. By H. F. Trewman. Isaac Pitman & Sons, London and New York, 1924. Cloth, 6 × 9 in., 244 pp., illus., diagrams tables, $7.50.

The object of the author is to present to serious students of railway engineering a number of the various problems encountered in electrification, with especial reference to the financial aspect. Discusses the manufacture and transmission of electricity, its conversion and distribution, railroad motors, energy consumption, suburban and trunk-line traffic.

LA RELATIVITÉ DÉGAGÉE D'HYPOTHÈSES MÉTAPHYSIQUES. By H. Varcollier. Gauthier-Villars et Cie, Paris, 1925. Paper, 6 × 10 in., 542 pp., 50 francs.

The object of the author is to disengage the work of Einstein from metaphysics and from the ideas of space and time that have become attached to it, and to give it the character of a new branch of analysis. Without contesting Einstein's results or even his theories, he attempts to show that these may be satisfactorily accounted for by a new theory based on our ordinary conceptions of space and time.

RIGHTS OF TRAINS. By Harry W. Forman. Simmons-Boardman Publishing Co., New York, 1925. (Railwaymen's handbook series.) Cloth, 5 × 8 in., 608 pp., illus., charts, $3.00.

The author discusses the Standard Code of rules, paragraph by paragraph. He explains the reasons for each rule and shows how it is to be interpreted and applied under various conditions. All this is done in a clear, direct manner which should appeal to railroad men.

CHARLES PROTEUS STEINMETZ, a Biography. By John Winthrop Hammond. Century Co., New York, 1924. Cloth, 5 × 8 in., illus., $4.

An authorized biography of the great engineer. Mr. Hammond follows his career from his birthplace in Poland to his position as principal consulting engineer to the General Electric Company, giving an agreeable portrait of the man and a readable account of his activities. Emphasis is placed on his social activities rather than on his technical achievements.

STRUCTURAL ENGINEERING, [vol. 2]; Fundamental Properties of Materials. By George Fillmore Swain. McGraw-Hill Book Co., New York, 1924. Cloth, 6 × 9 in., 200 pp., illus., diagrams, tables, $2.50.

This, the second volume of Dr. Swain's treatise in structural engineering, is not intended to replace detailed works on materials nor to describe processes of manufacture. It is intended to bring together in logical fashion the information needed by the constructing engineer concerning the fundamental properties of the principal materials, their constitutions and physical structure, the importance and effect of various ingredients, the effect of different treatments and the significance of the specifications used to secure sired properties in them.

TALE OF OUR MERCHANT SHIPS. By Charles E. Cartwright. E. P. Dutton & Co., New York, 1924. Cloth, 5 × 8 in., 275 pp., illus., $3.

A popular account of the maritime activities of the United States from their beginning to the present day. Traces the evolution of merchant vessels and of rigs. Illustrated with attractive drawings by the author.

TESTING OF HIGH-SPEED INTERNAL-COMBUSTION ENGINES. By Arthur W. Judge. D. Van Nostrand Co., New York, 1925. Cloth, 6 × 9 in., 392 pp., illus., diagrams, tables, $7.50.

As a companion to his previous works on internal-combustion engines, Mr. Judge has now issued this work on the methods of testing these engines and the apparatus used in testing. He has tried to present the subject in a practical way, neither too elementary nor too advanced for the ordinary needs of engineers and manufacturers, while the needs of those who are engaged in advanced work and in research are met by references to original papers and by bibliographies.

THERMODYNAMIQUE. By Aimé Witz. Fourth edition. Gauthier-Villars et Cie., Paris, 1924. Paper, 5 × 8 in., 333 pp., 20 fr.

A brief statement of those general principles of thermodynamics which an engineer needs to know, presented in rational order and arranged for convenient reference. The book will prove useful as an introduction to the study of heat engines in general and to the theory of steam and gas engines. The present edition has been revised and extended.

UEBER DIESEL-ELEKTRISCHE LOKOMOTIVEN IM VOLLBAHNBETRIEB. By Herbert Brown. Ernst Waldmann, Zürich, 1924. Paper, 6 × 9 in., 73 pp., diagrams, tables.

The Diesel engine is, the author of this work says, the most economical heat engine of today. It is therefore very desirable that it be also used for locomotives. The problem has been solved for light locomotives but not for those for trunk lines, and the purpose of the present study is to test suitability of the Diesel engine for heavy locomotive work.

In part 1 the pamphlet reviews briefly the various kinds of Diesel locomotives which have been proposed. The remainder of the book is concerned with exact investigations of the Diesel-electric locomotive, the type most favored at present. This type is examined theoretically and practically, from both a technical and an economic viewpoint.

VUES GÉNÉRALES SUR LA THEORIE DE LA RELATIVITÉ. By A. S. Eddington. Translated and edited by Thomas Greenwood: Gauthier-Villars et Cie., Paris, 1924. Paper, 5 × 8 in., 103 pp., 10 francs.

The articles here collected and issued in French translation have appeared at various times between 1918 and 1922 in leading scientific journals. They present an outline of Professor Eddington's views in convenient form and should prove valuable to students of the subject, especially in France.

DAS WIRTSCHAFTLICHE AMERIKA. By Carl Köttgen. V. D. I. Verlag, Berlin, 1925. Cloth, 6 × 8 in., 178 pp., illus., charts, 5 marks.

In 1924 this writer traveled in the United States for the purpose of studying economic conditions and particularly to determine the effect that American conditions might have on the industrial development of Germany. His book briefly presents his findings with respect to our agricultural, mining and manufacturing industries; the condition of the worker, national wealth, etc. and compares the situation here with that in Germany.

DIE ZEITGEMÄSSE HEISSDAMPFLOKOMOTIVE. By Robert Garbe. Julius Springer, Berlin, 1924. Boards, 8 × 11 in., 167 pp., illus., diagrams, tables, $3.35.

This treatise discusses the development of the locomotive in recent times, with particular reference to the use of superheating. The author discusses investigations on the relations between grate surfaces and heating surfaces, the combustion phenomena in the firebox, methods of firing, and the efficient production and utilization of superheated steam. From these investigations he draws conclusions concerning the design and operation of freight and passenger locomotives and illustrates his opinions by applying them to the design of an actual locomotive.